AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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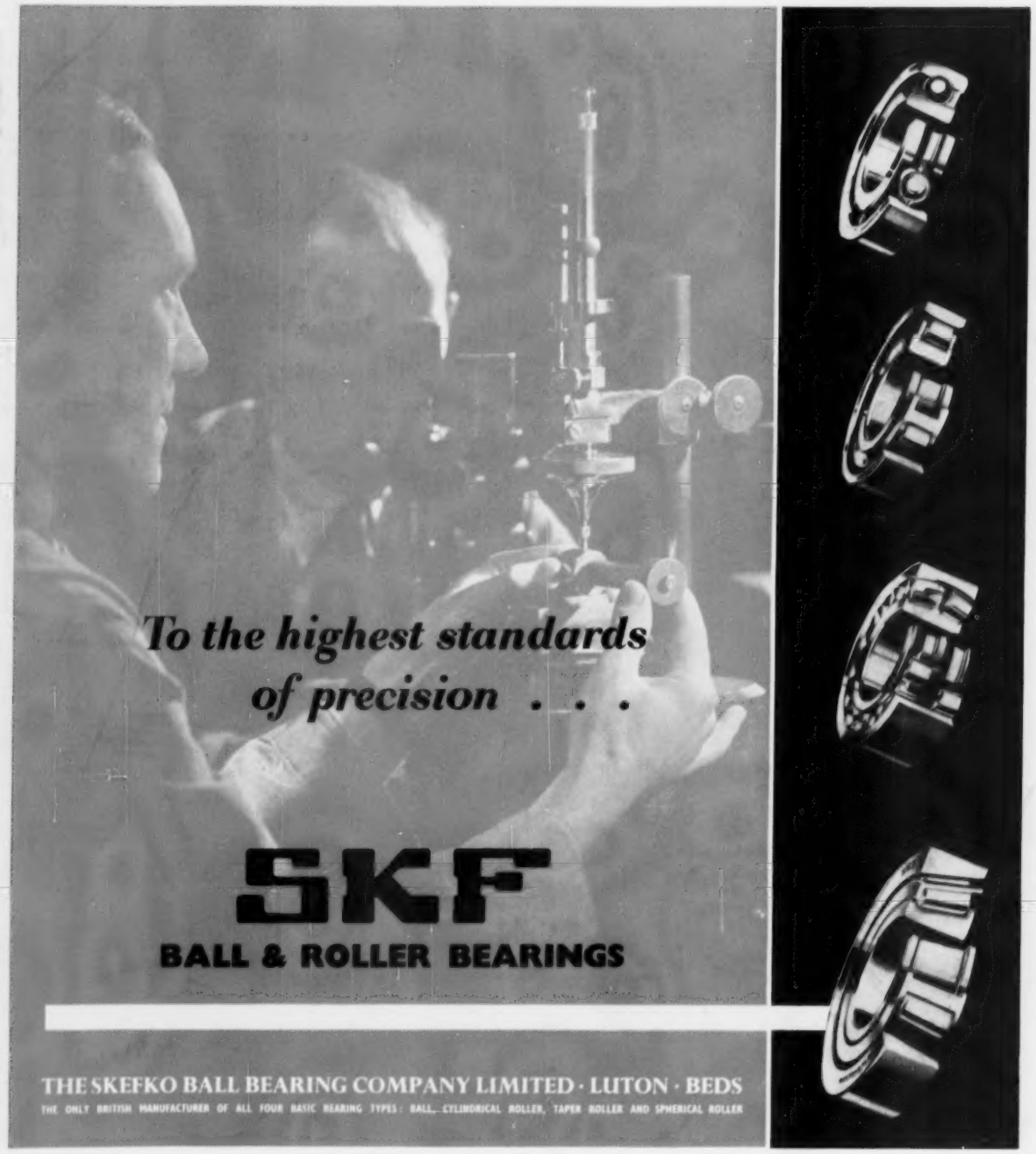
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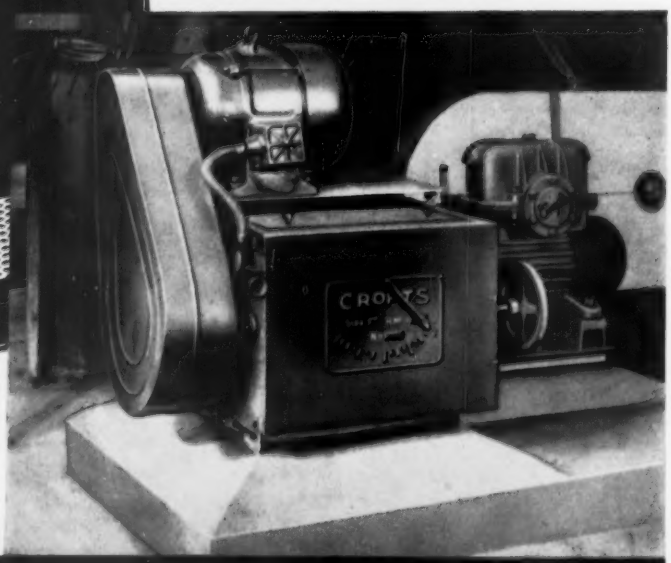

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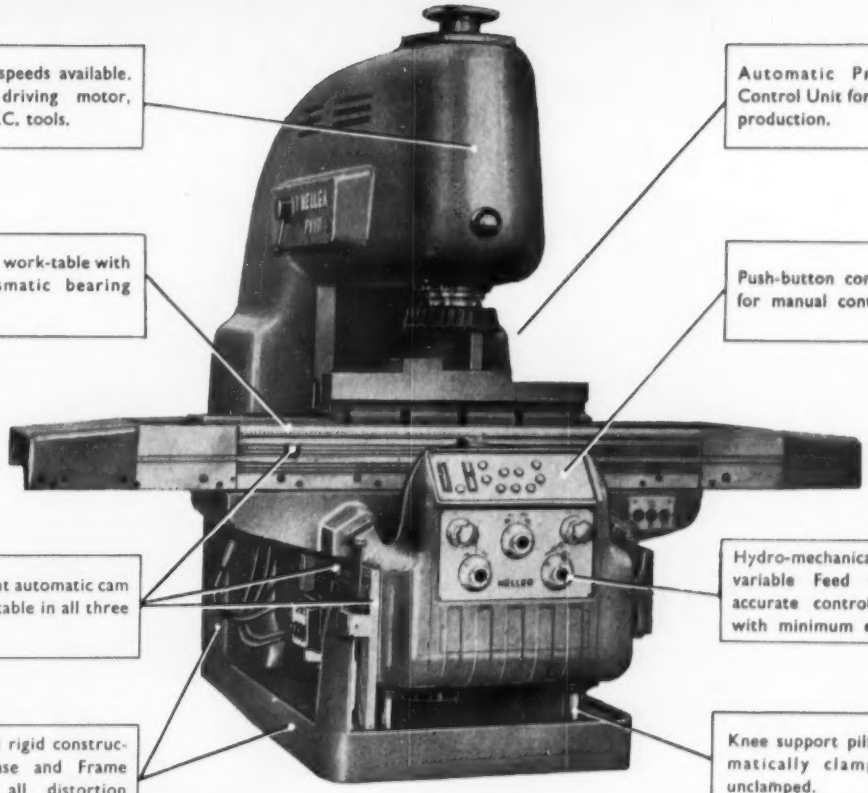

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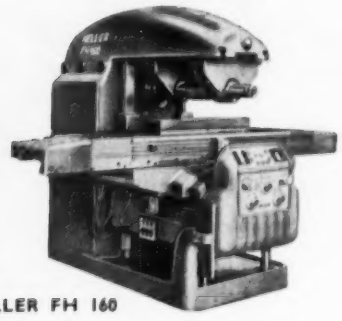

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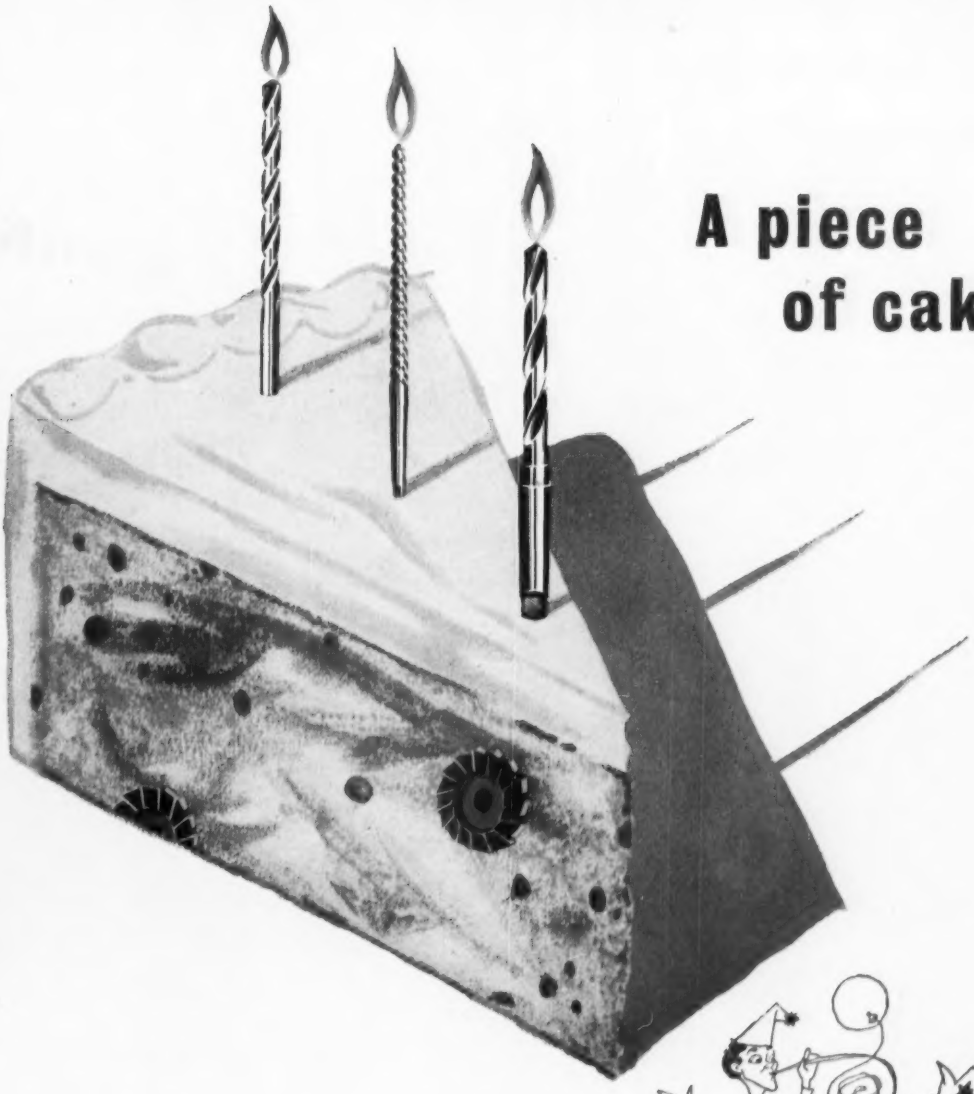

At first sight, wholly irrelevant, but those of us with growing sons know, only too well, the full import of the phrase . . . 'O.K. Dad —it's a piece of cake!' It implies, at once, absolute confidence, a certain facility for doing a job quickly and well. In other words . . . 'it's as good as done!' There's nothing more to add, is there?

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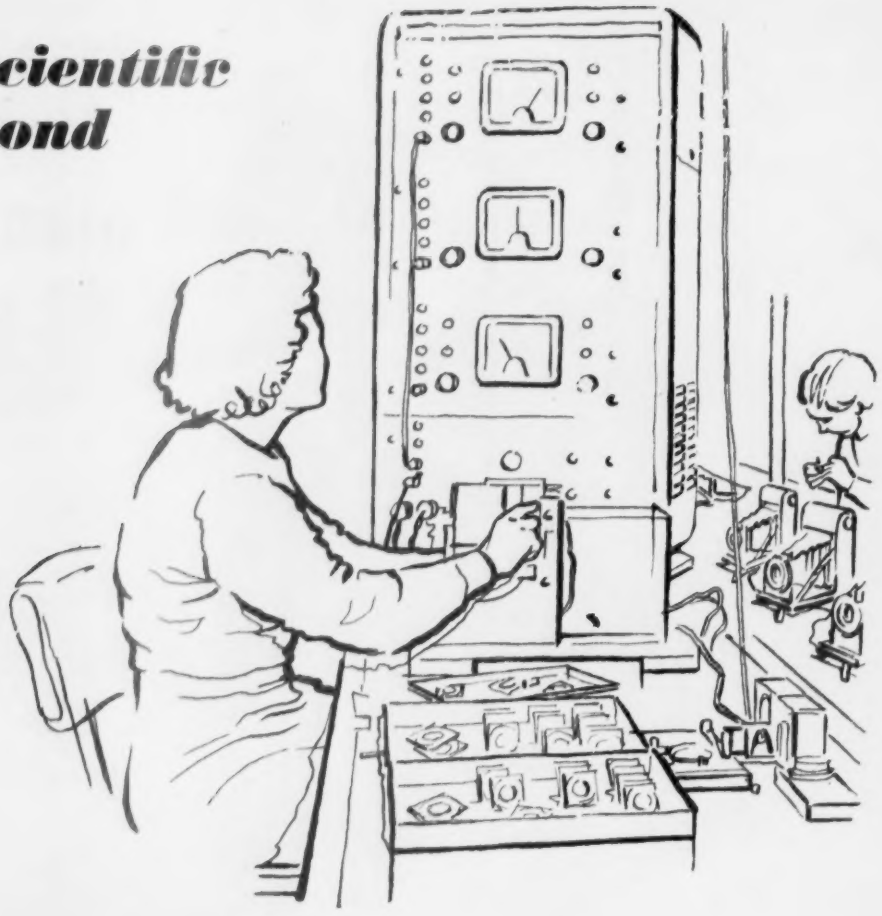

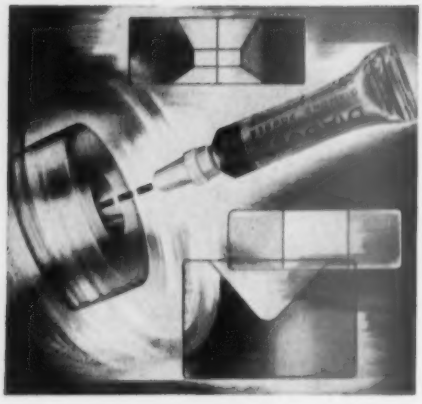

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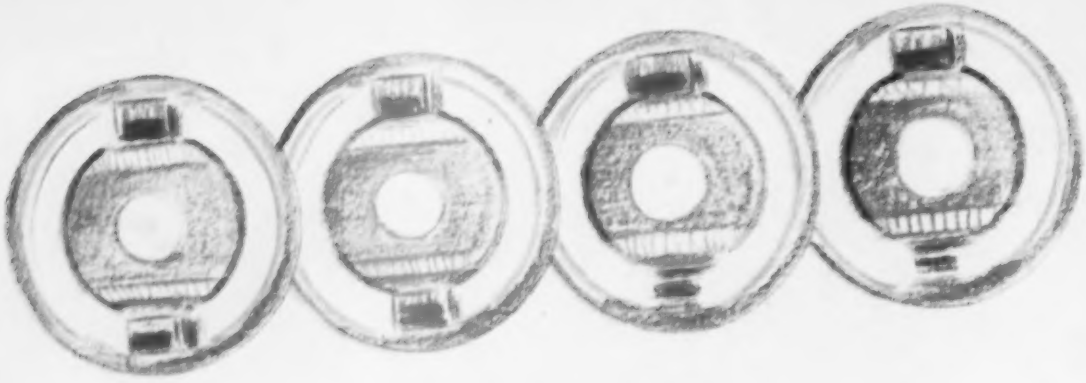
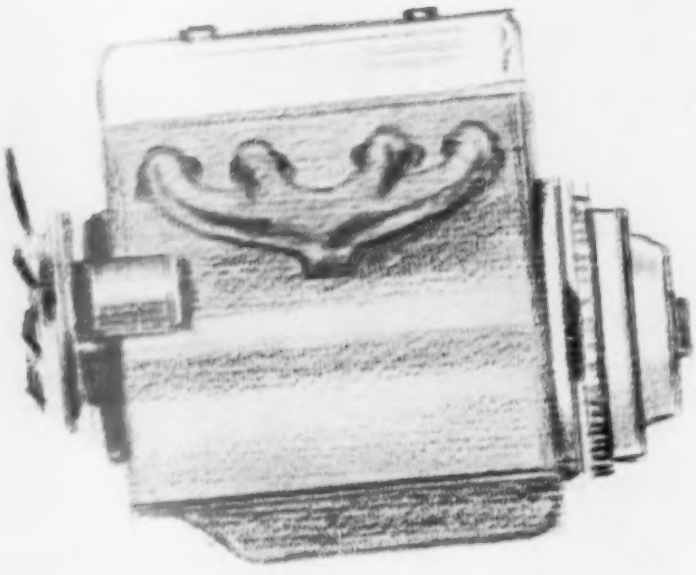


equal five times:-



A car may have a good acceleration but the brakes will stop it - from a given speed - in a fraction of the time taken to reach that speed.

In other words, the brakes are handling 4 or 5 times the energy provided by the engine in accelerating the car.

BRAKES, viewed as heat engines

Since the majority of brakes fitted to road vehicles are bought from specialists such as ourselves, there is a tendency to regard them as being like so many other items, components which function merely as pieces of mechanism. In order to understand the problems of brake design and performance, it is desirable that this outlook should be modified. Brakes must be considered as being a form of heat engine having a heavy-duty task to perform, since, under conditions almost unpredictable, and often based upon conjectures which are almost improbable, they nevertheless perform the nearly impossible.

They are not, of course, true heat engines in the benevolent sense, because they are not constructive translators of heat into useful work, or work into useful heat. On the other hand, they are simply there as 'negative' heat engines, to dissipate mechanical energy, just as a refrigerator is there to dissipate heat.

The amount of energy handled by brakes is not generally realized. Whereas an engine—massive, heavy and complex, the subject of great precision in its essential parts and always accompanied by a host of ancillaries—gives out the power which, on the level and in not so short a time, builds up the momentum of the car, the brakes, though compact, relatively light and simple units, are on the other hand expected to check that momentum in a fraction of that time.

As a result, the energy passing through the brakes is the equivalent of many times the horse-power developed by the engine. In present-day brakes, and probably in brakes for many years to come, that energy is dispersed in heat, yet under conditions which are almost antagonistic to its dissipation.

The energy dissipated in six crash stops by a fast car is equivalent to the energy required to lift a man a distance equal to the height of Mt. Everest above sea-level.

The problems arising out of this rapid heat dissipation are many.

So far as present practice is concerned, the first consideration is to get plenty of air to the brakes, thus they are cooled by forced convection plus a certain amount of radiation.

But when the car is at rest after a crash stop, the forced convection is replaced—in the absence of the airstream—by a much less efficient convection, so that the rest of the cooling is effected by radiation, conduction being negligible, and a second brake application may find the brake still hot.

Many brakes must owe their very survival to the obliging fact that the rate of radiation goes up so rapidly in relation to temperature, for a rise of 20% in absolute temperature relative to the surroundings doubles the amount of heat radiated.

What we need, in addition to good cooling, is some kind of 'reservoir' which will absorb some of the heat and dissipate it later.

The greater the effectiveness of these processes, the lower the temperature of the brake surface and brake lining, two most desirable factors, since they influence the whole efficiency of the brake under strenuous conditions, and, of course, its durability.

In some later essays, factors of design, and some others, will be discussed.

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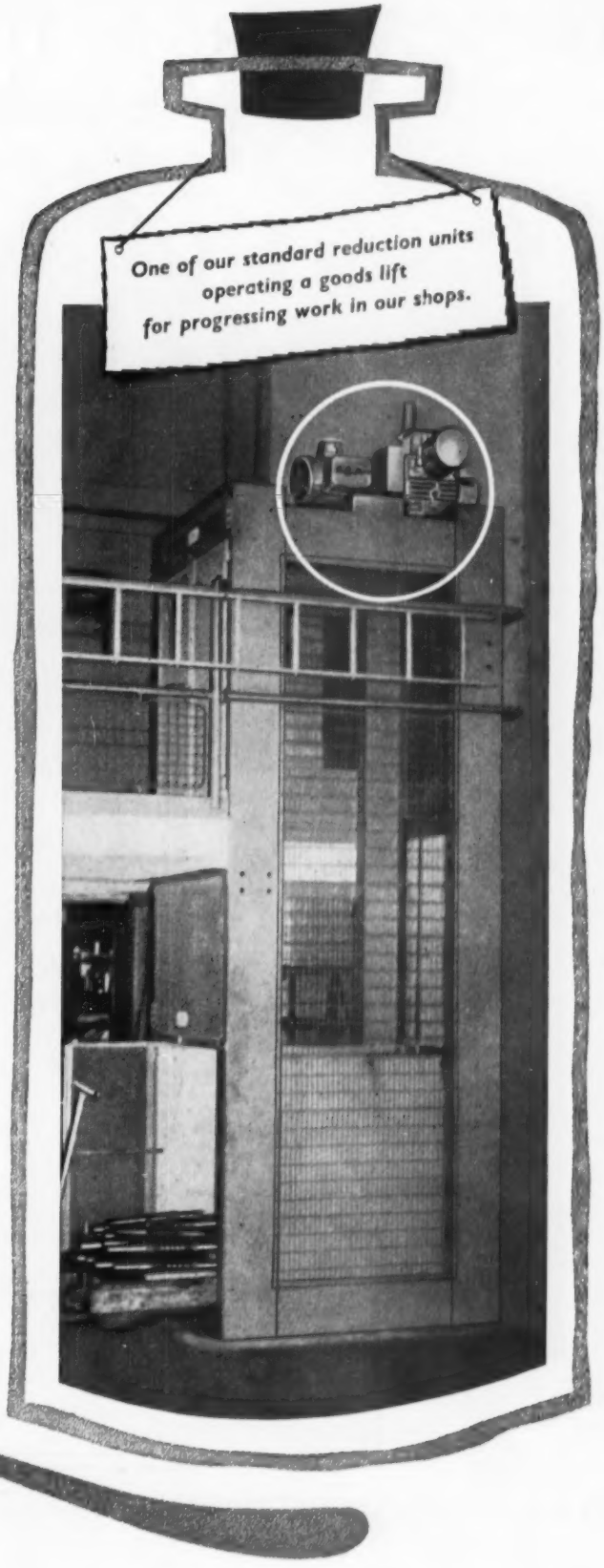

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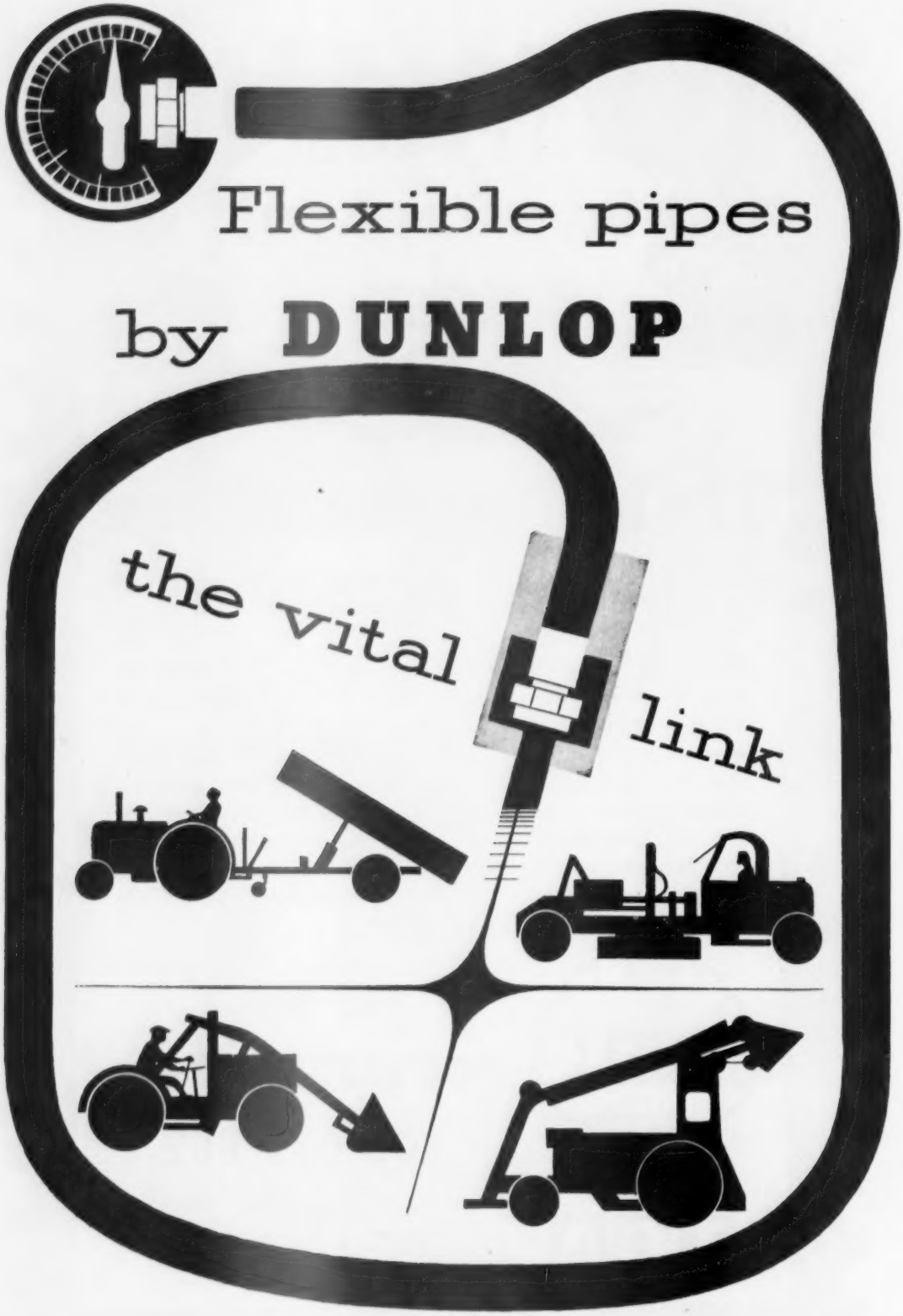

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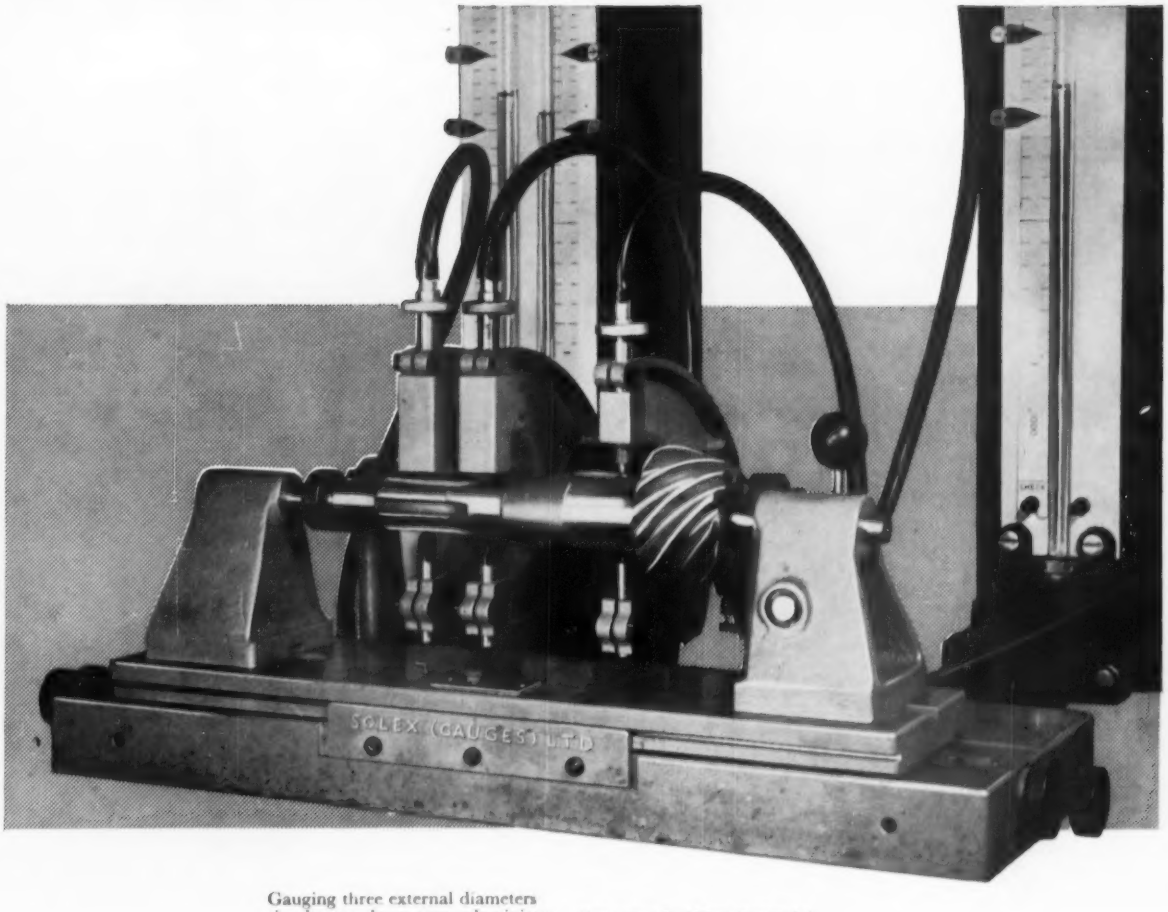

Gauging three external diameters simultaneously on rear axle pinions. *Photo courtesy Ford Motor Company Ltd.*

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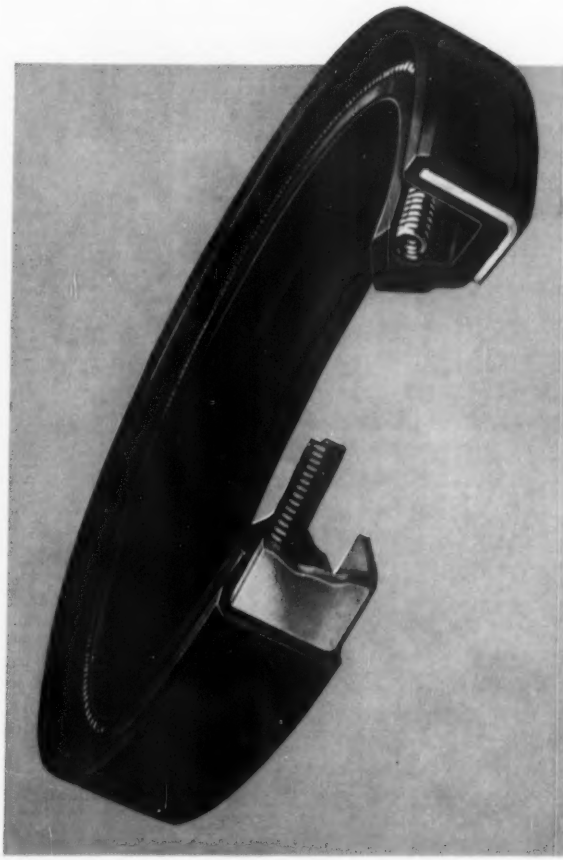

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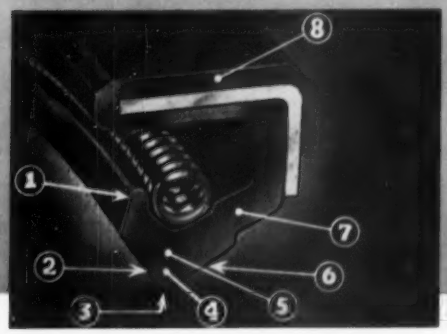

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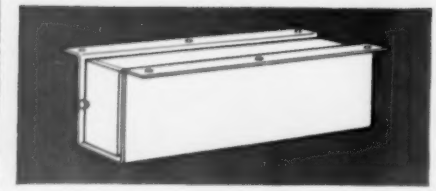

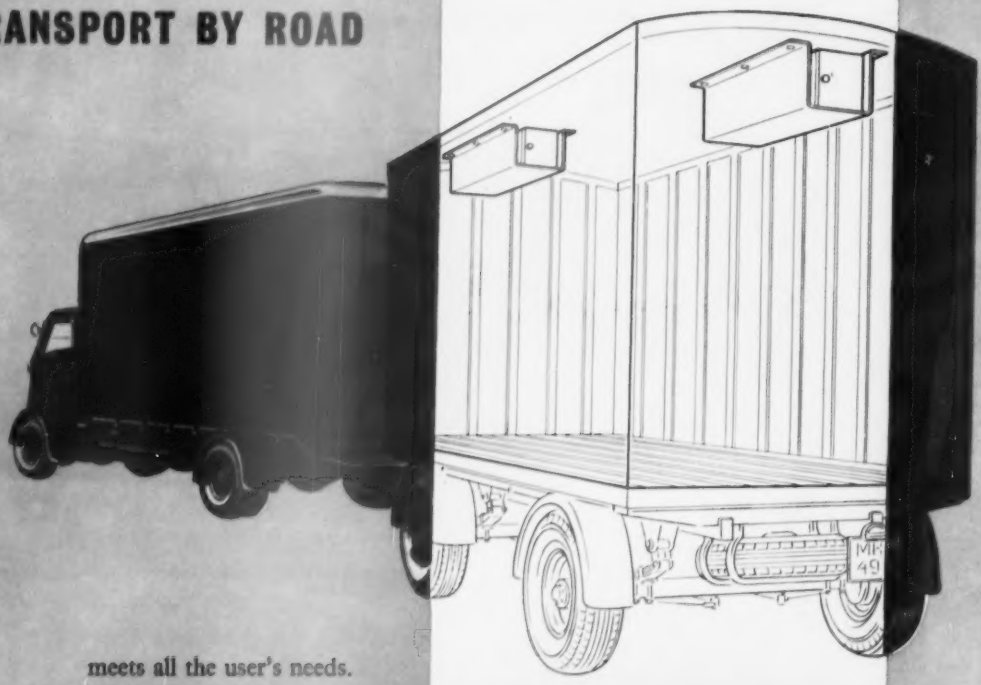

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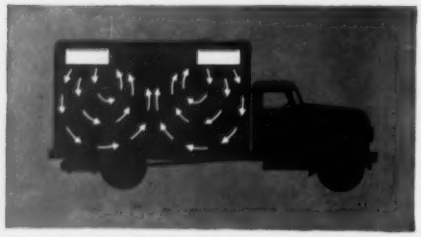

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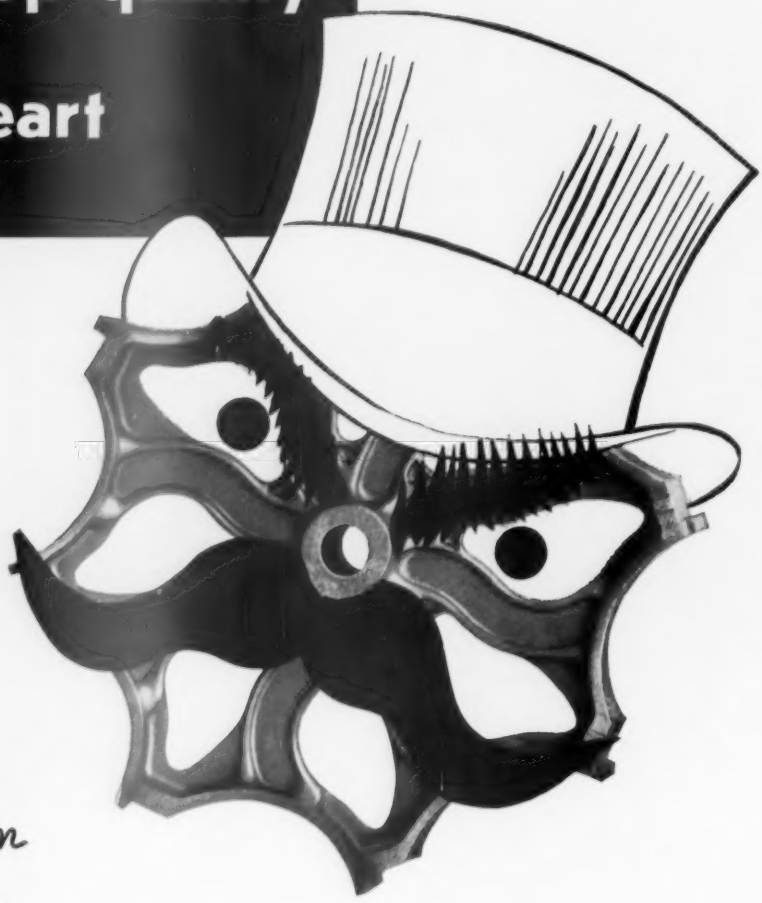

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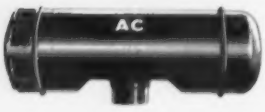

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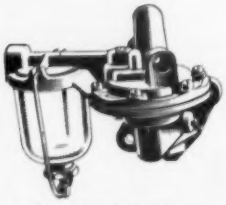

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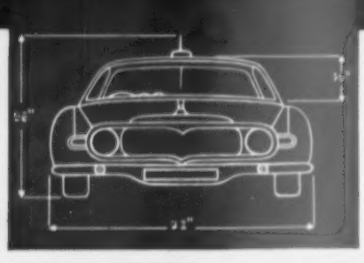

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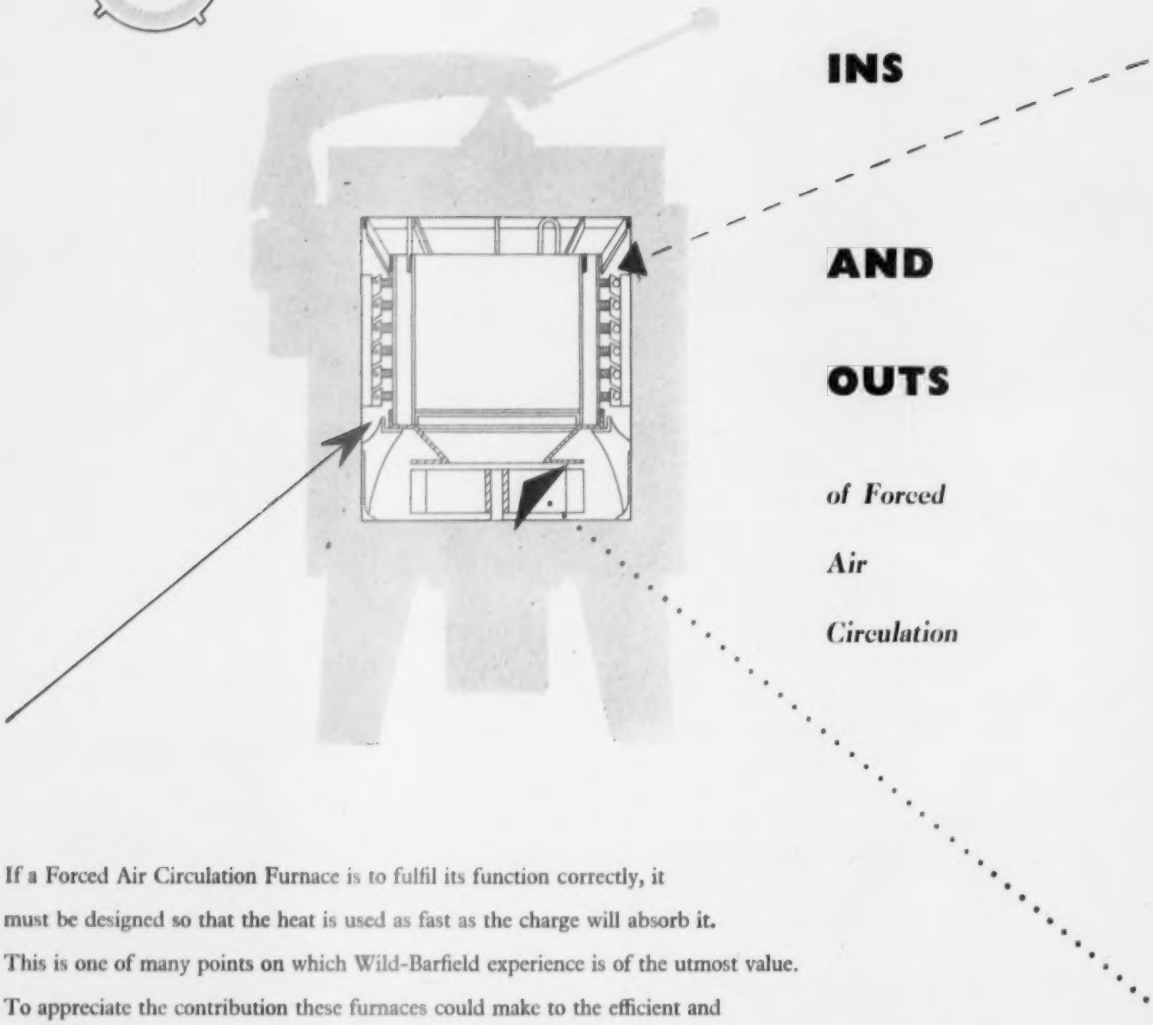

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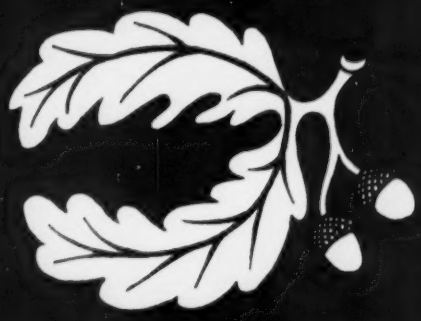

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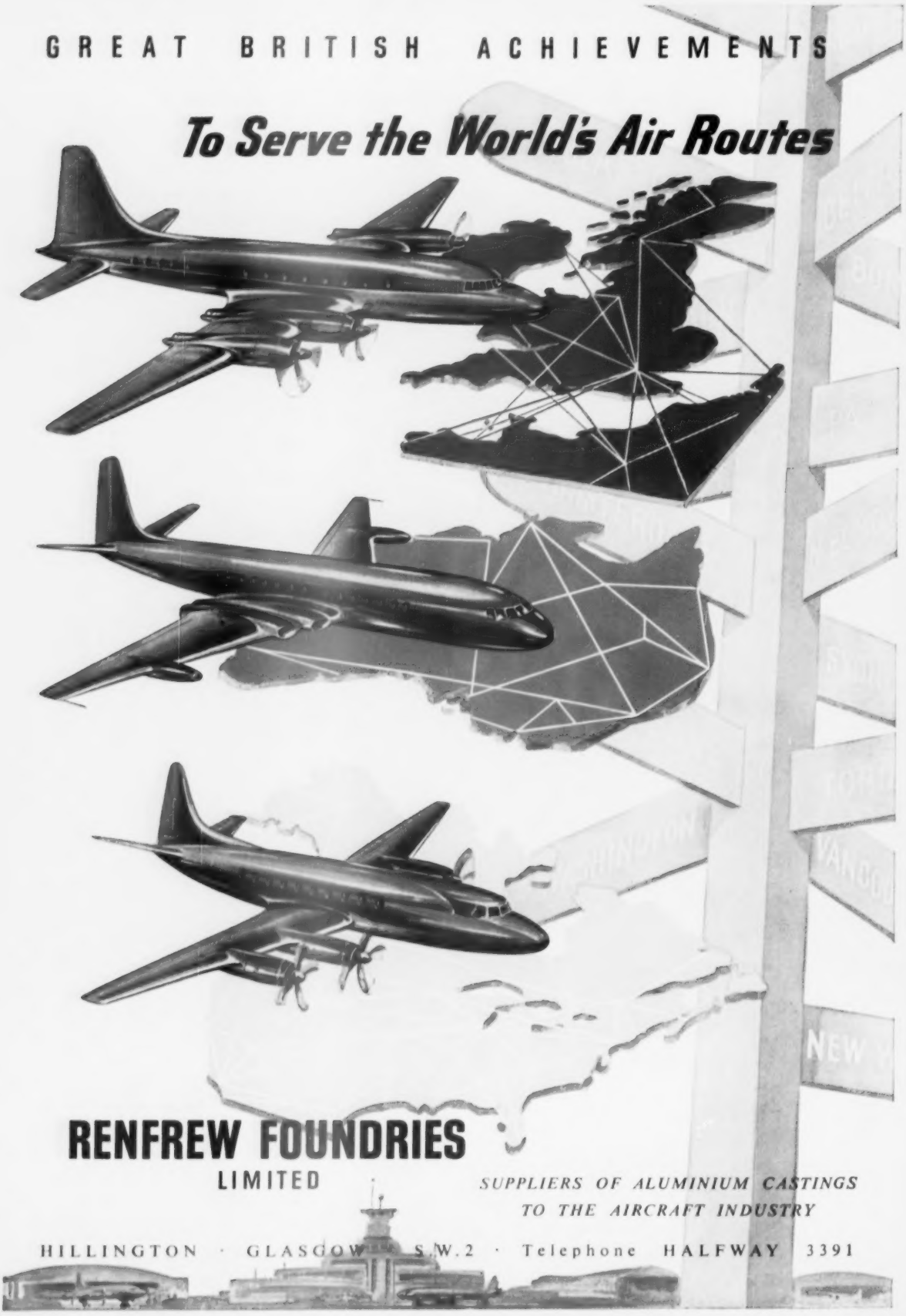

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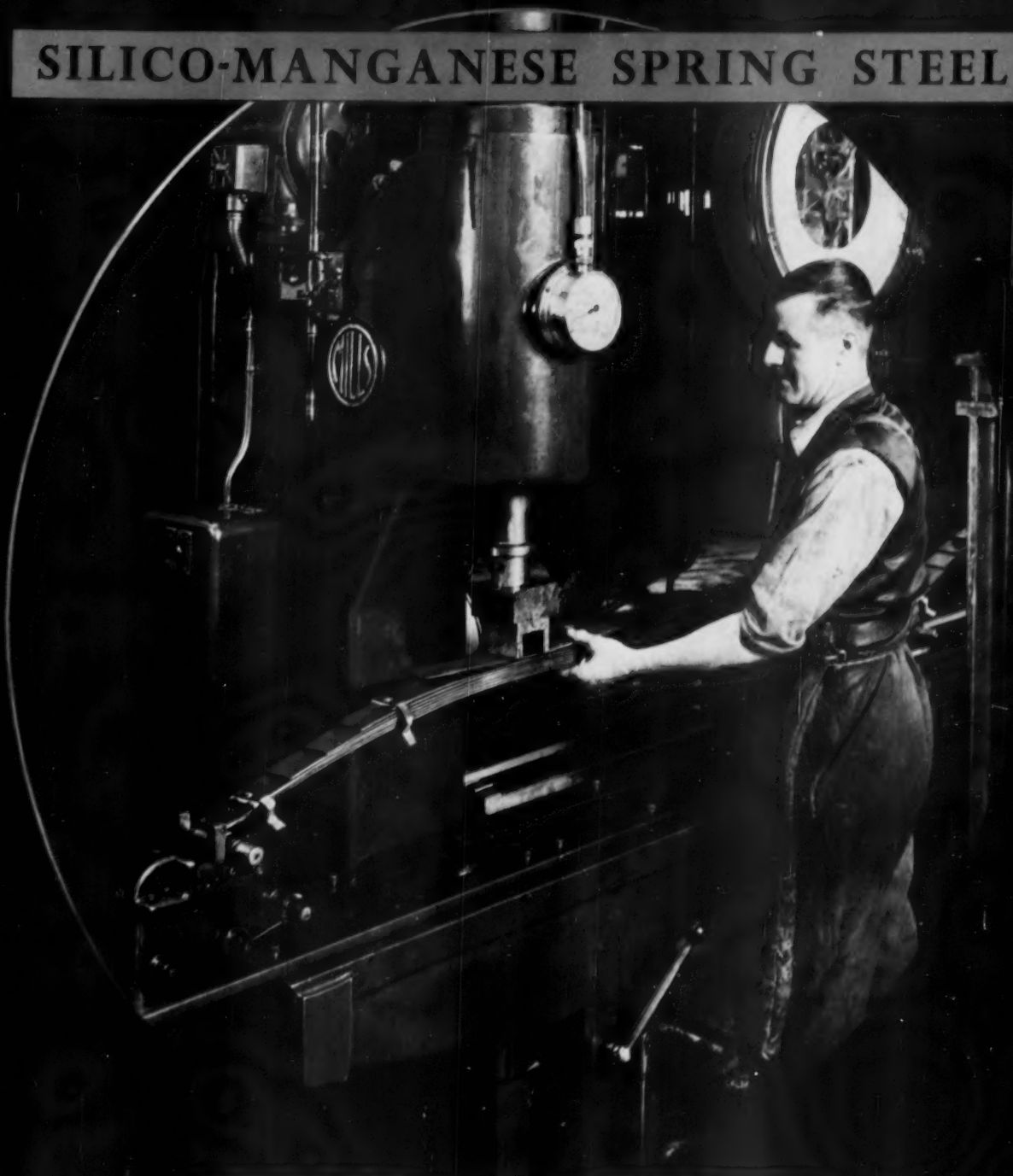

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Since its introduction to the British Motor Industry five years ago, the Laycock-de Normanville Overdrive has firmly established itself as the only really satisfactory Overdrive for British Cars. This is evident by the fact that it is now available on 22 of Britain's finest cars.

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The Laycock-de Normanville Overdrive is available on the following cars :— Austin Healey, Bristol 405, Doretti, Humber Hawk, Jaguar Type 'M', Jaguar XK140, Jensen Interceptor and 541, Sunbeam Alpine, Standard Vanguard and Vanguard Diesel, Sunbeam Mk. III, Triumph Renown, Triumph T.R.2.

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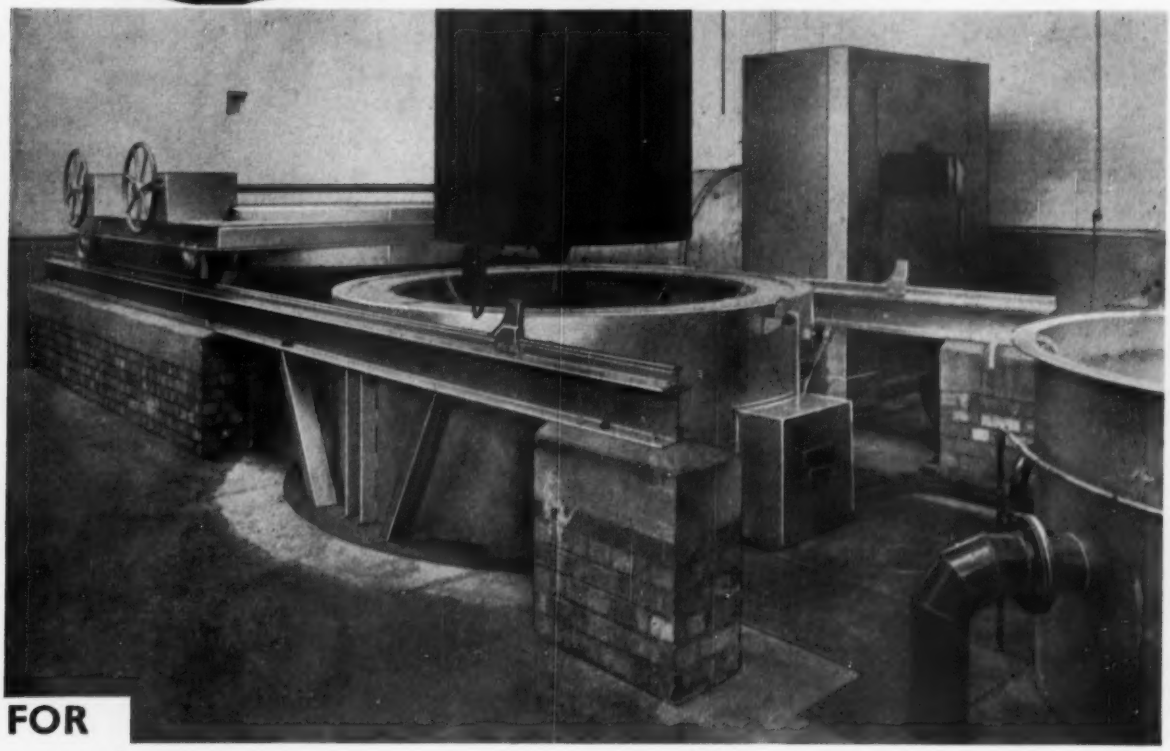

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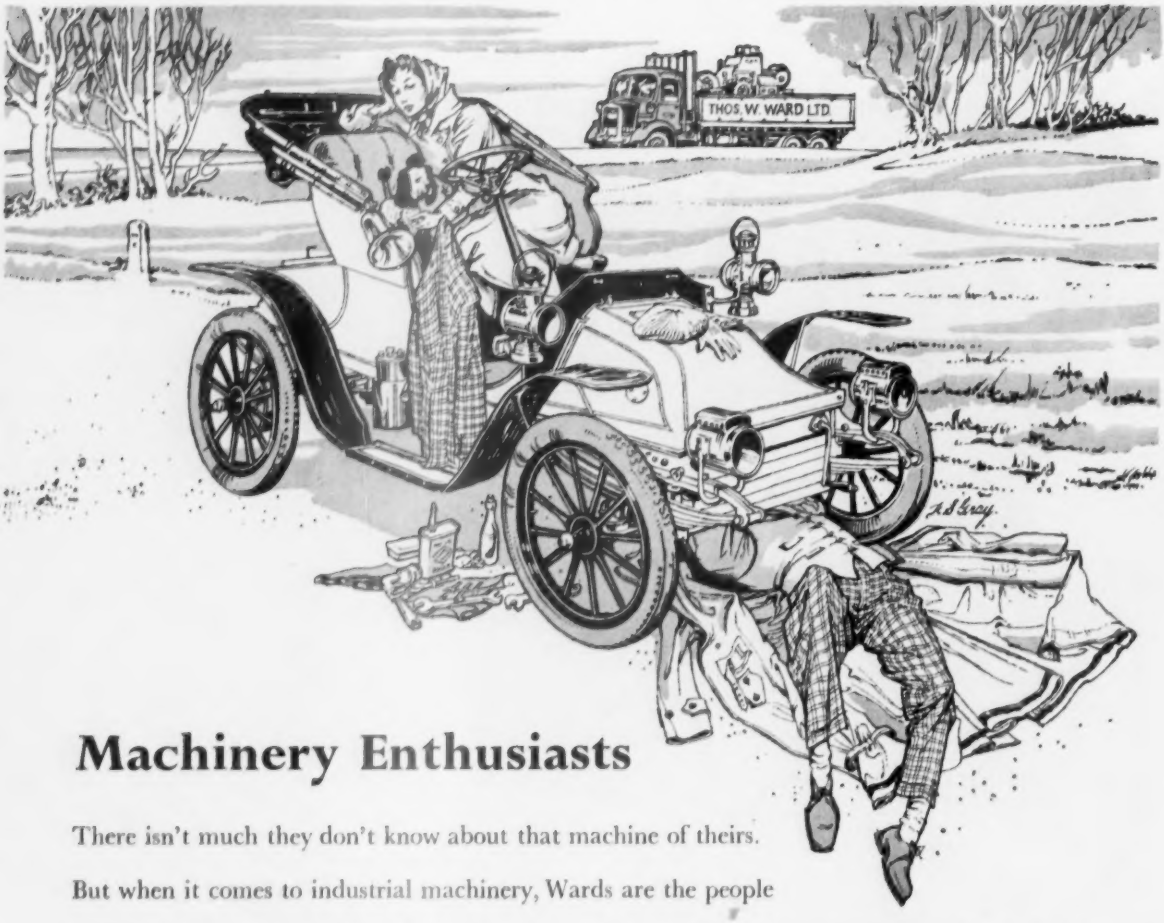

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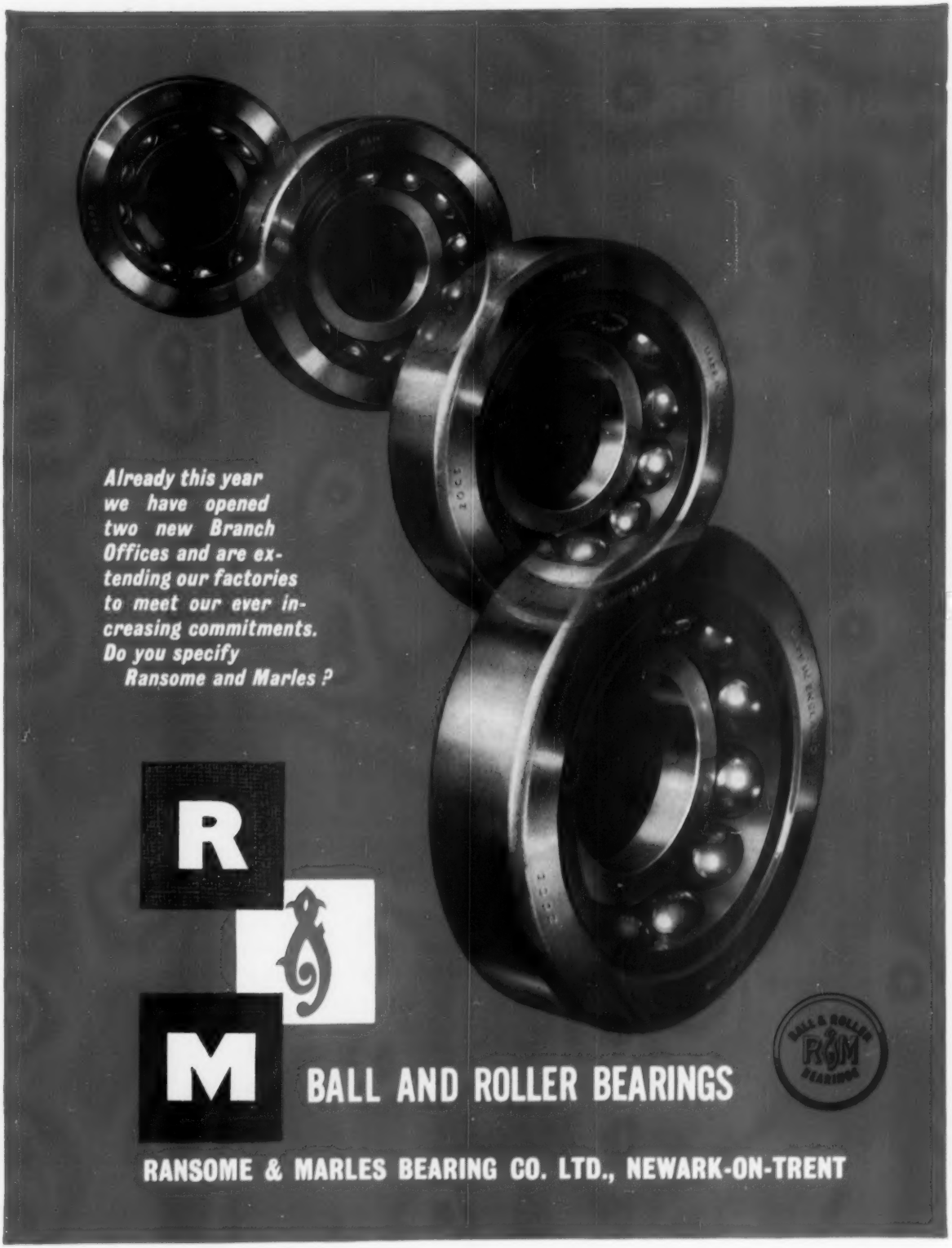
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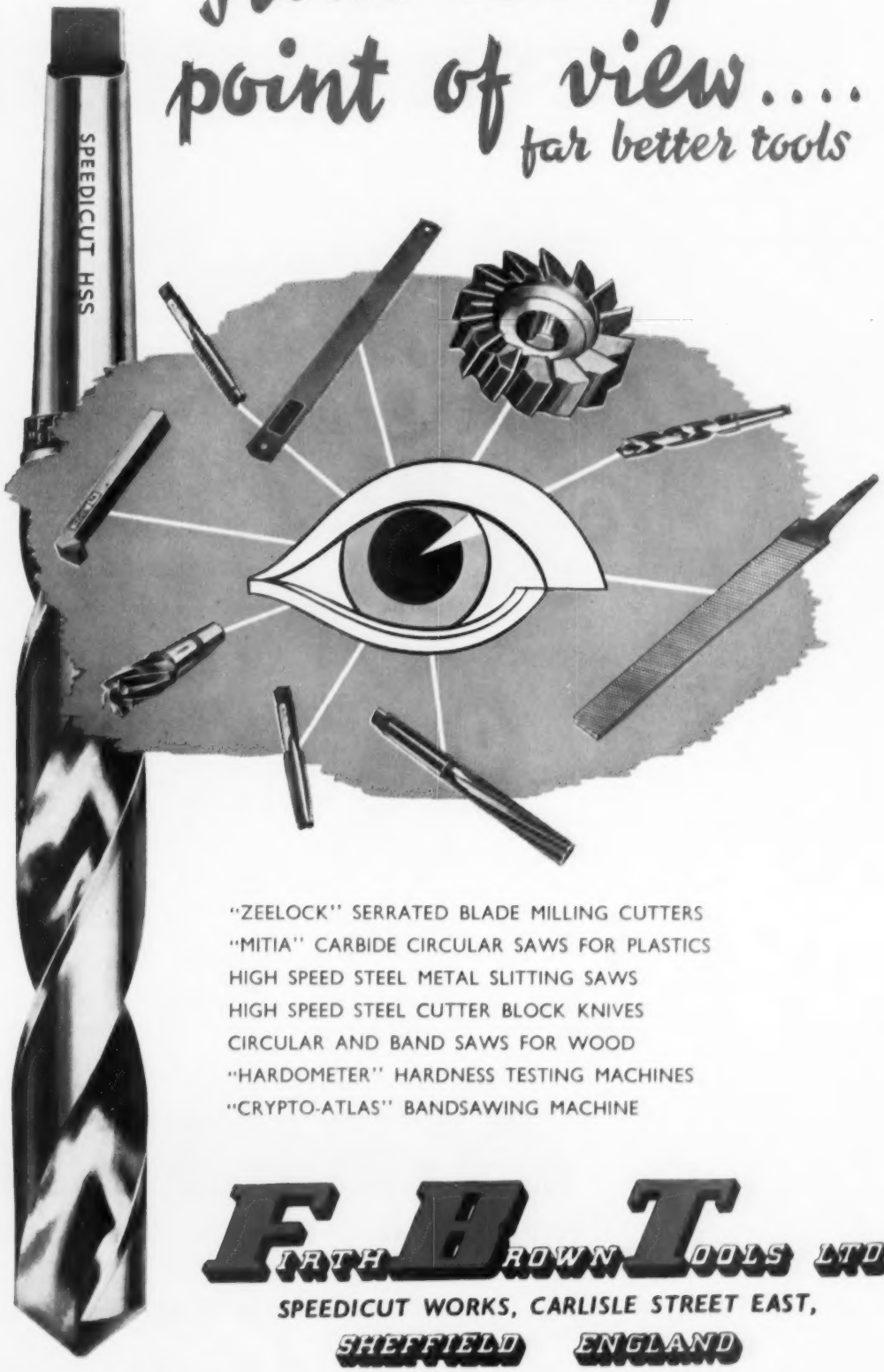

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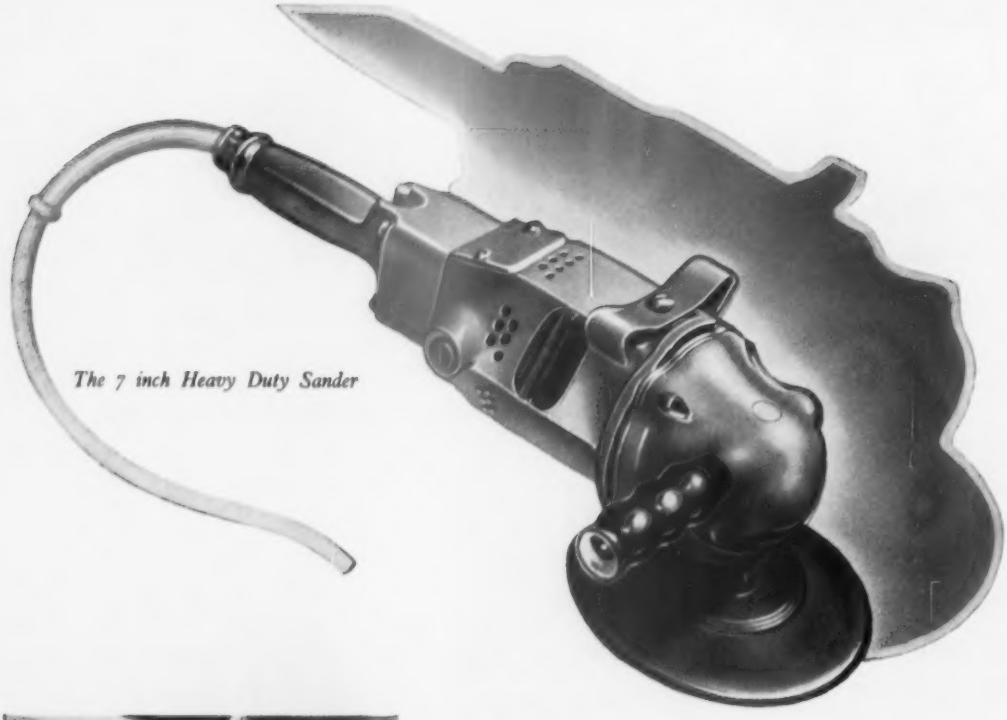

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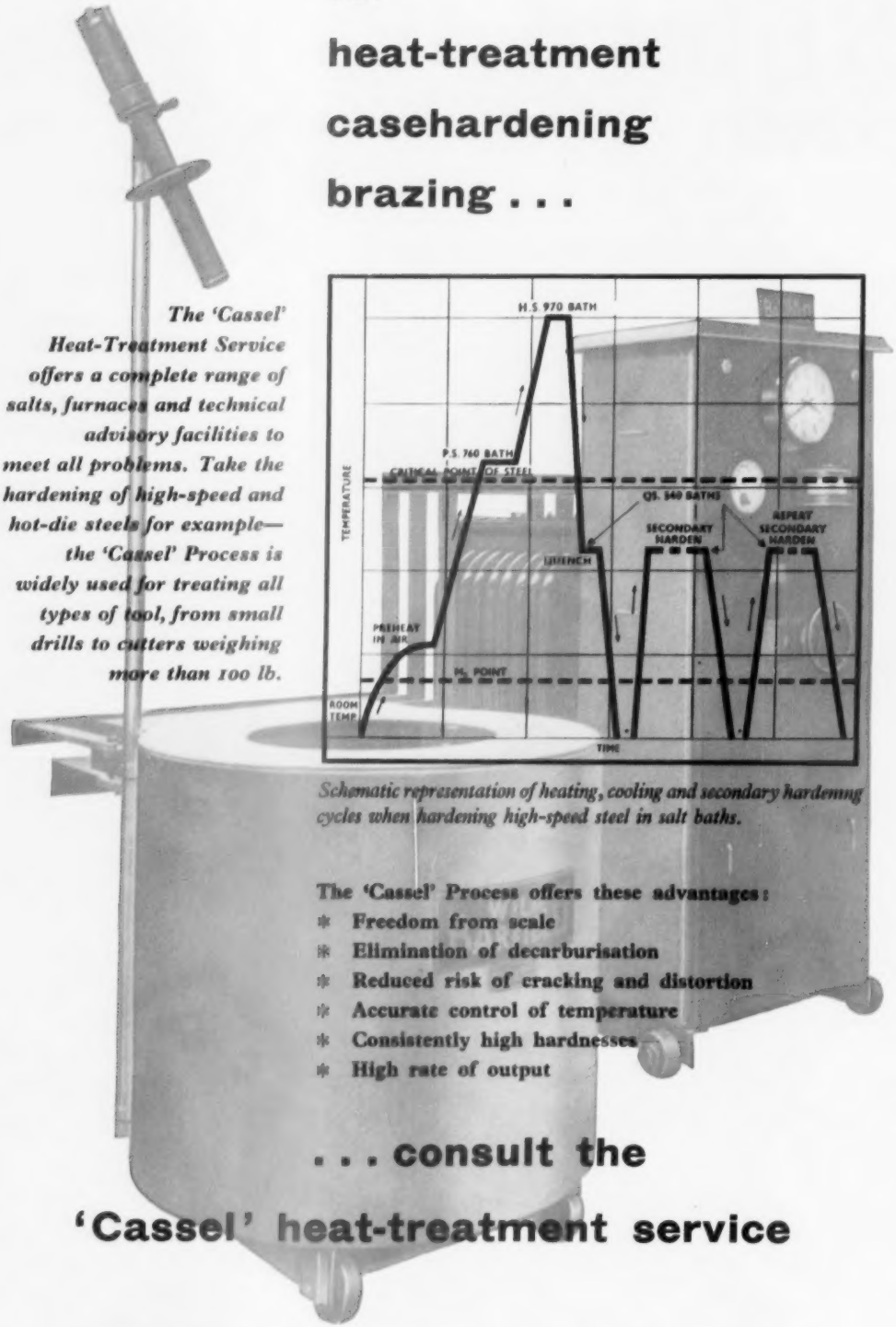

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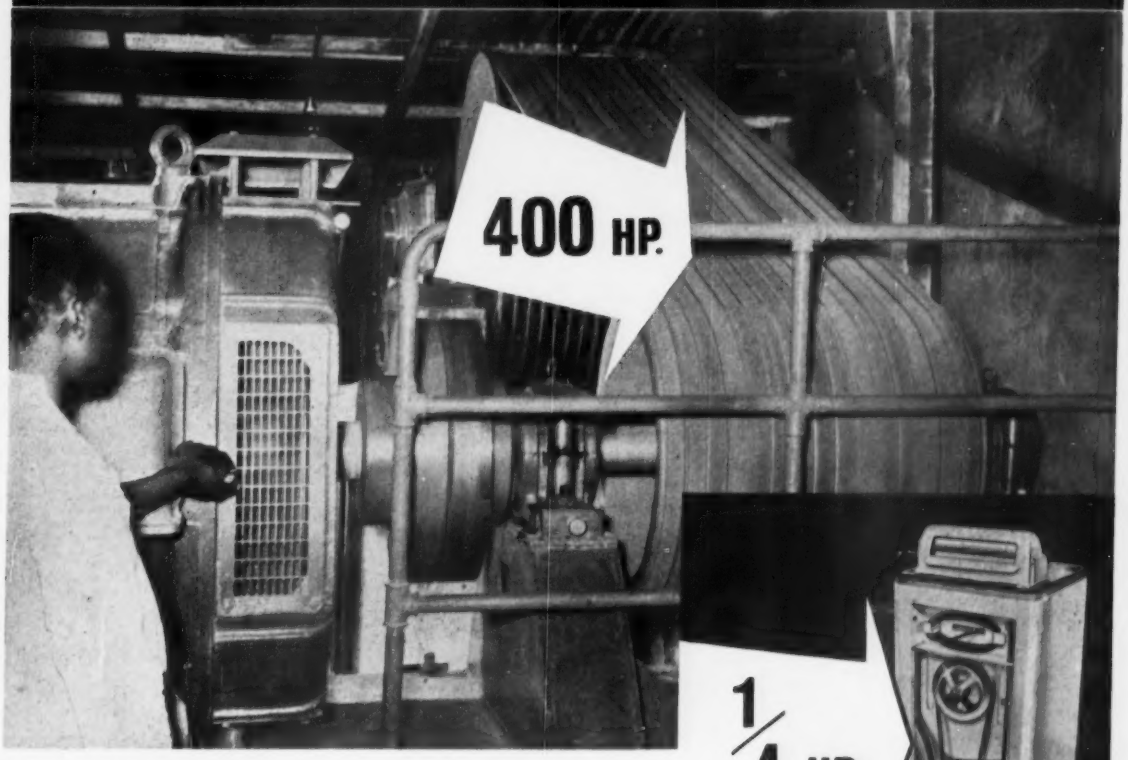

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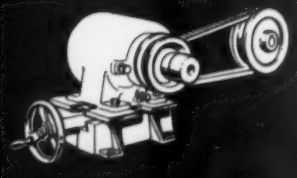

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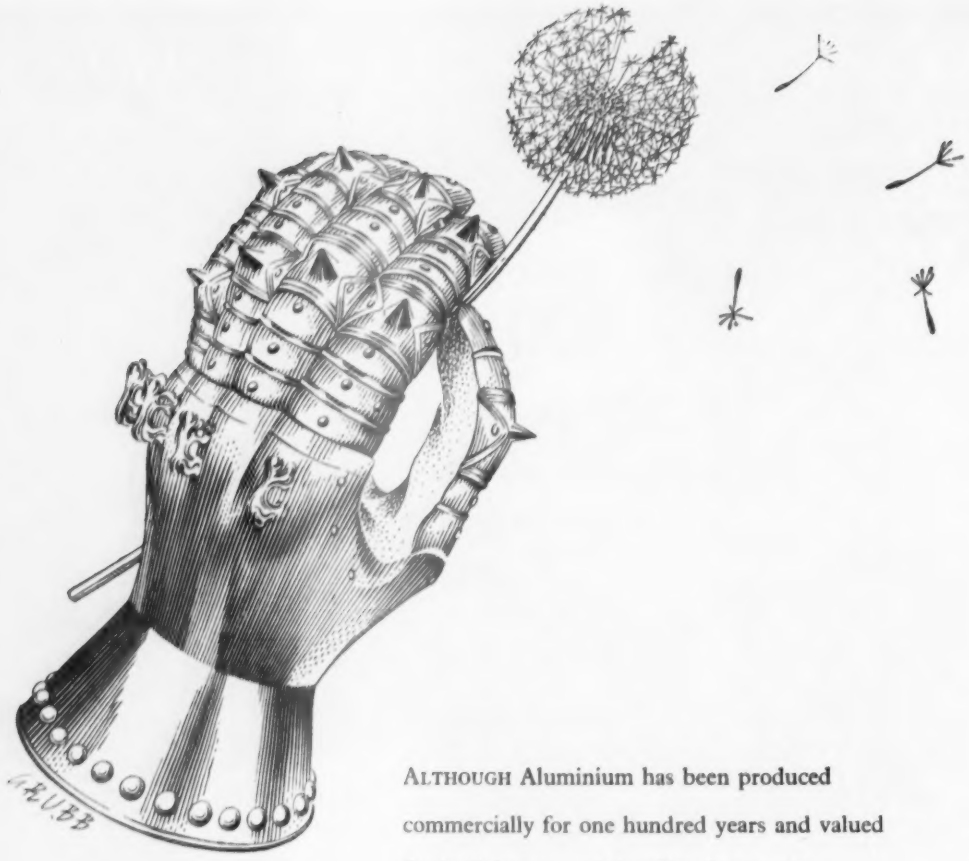

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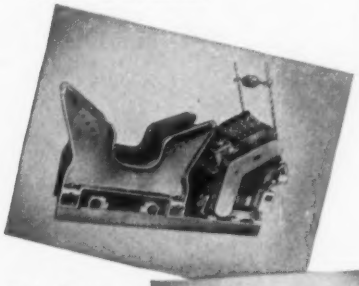
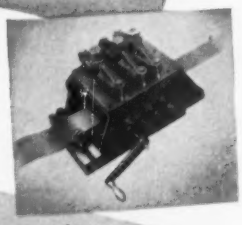


Plain Stock Straightener

For use with Slide Feeds—range from 2½" to 12" wide.



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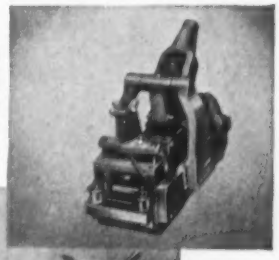
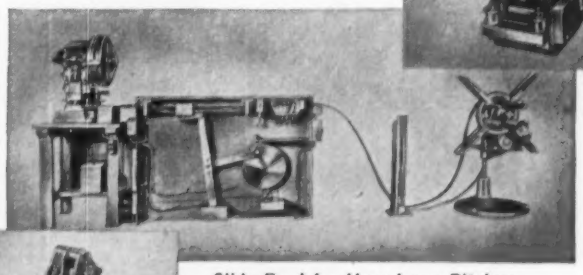
Makes press fully automatic. Available up to 20" wide.





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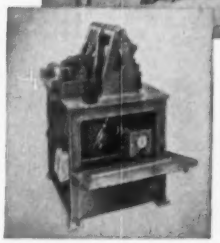
Feeds any preset length of material automatically.



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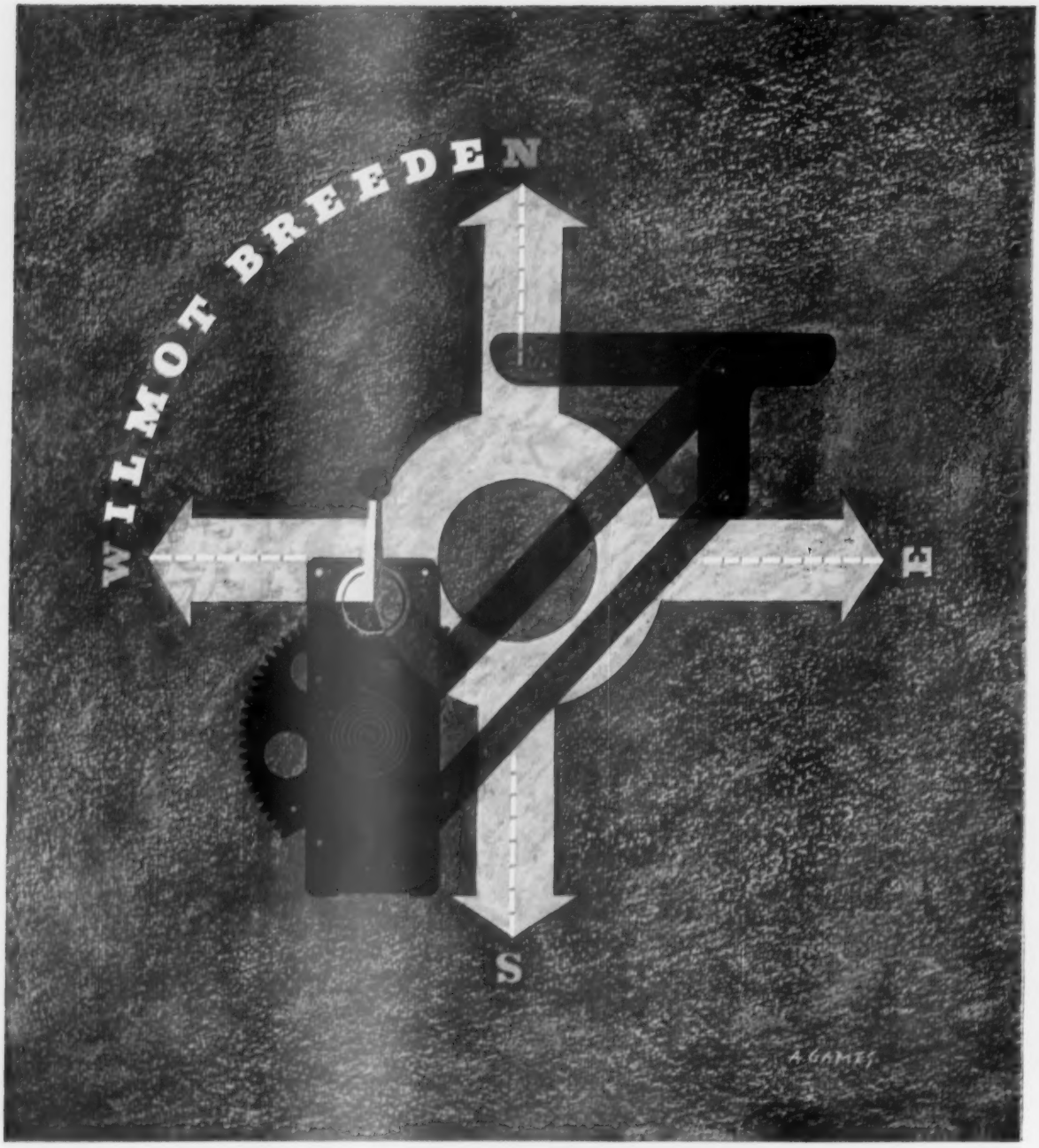

across the world millions of car windows are controlled by Wilmot Breeden window regulators. Yet because they give lasting, undemanding service many motorists scarcely know what they look like.

The parallel arm regulator in the drawing consists essentially of a pair of spur gears, a coil spring to counterbalance the glass, an automatic device to lock the window at any height and two parallel arms to keep it level. This simple and efficient mechanism is the product of sound engineering practice both in design and manufacture.

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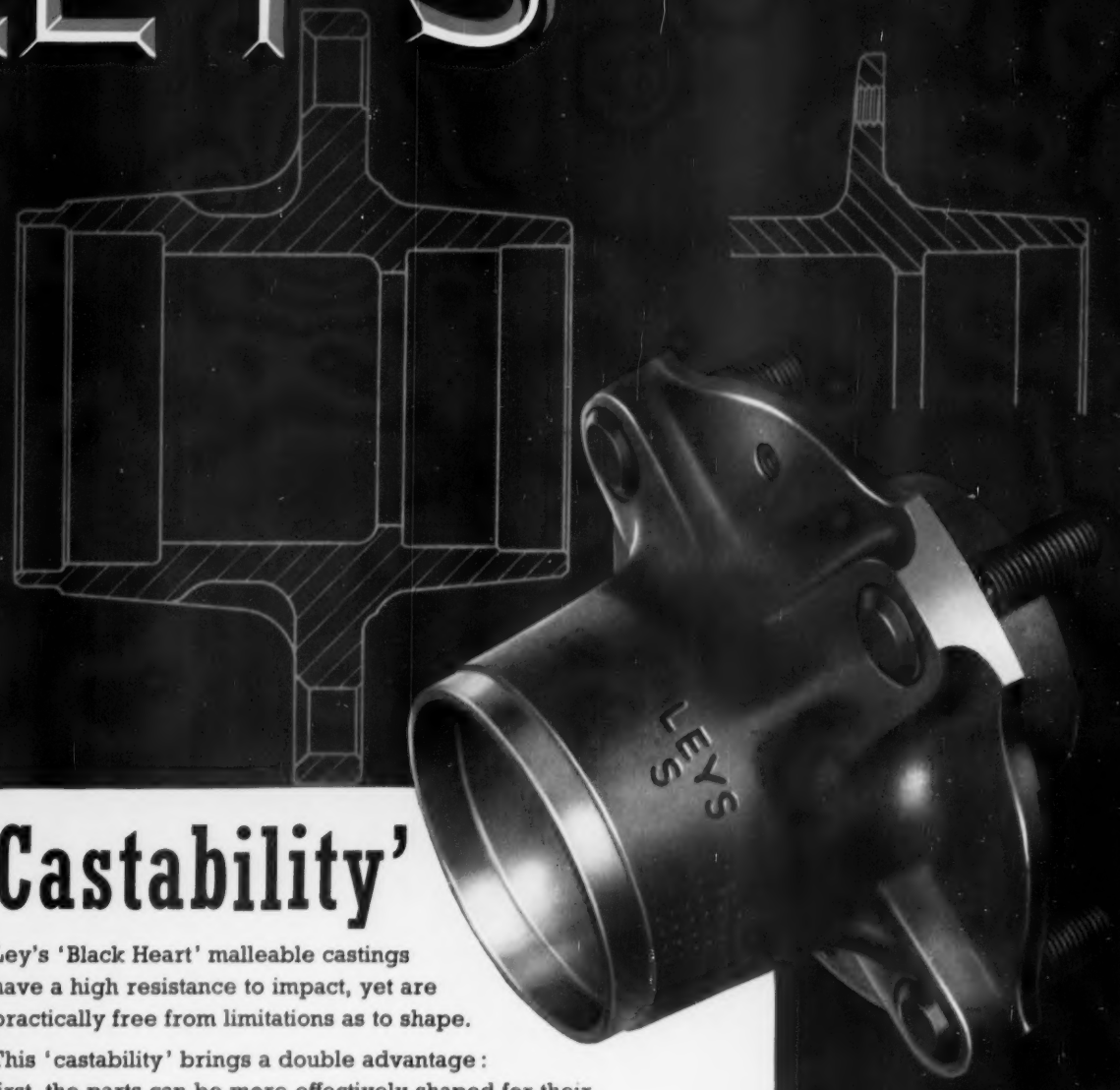

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We offer designers the full collaboration of our engineers.

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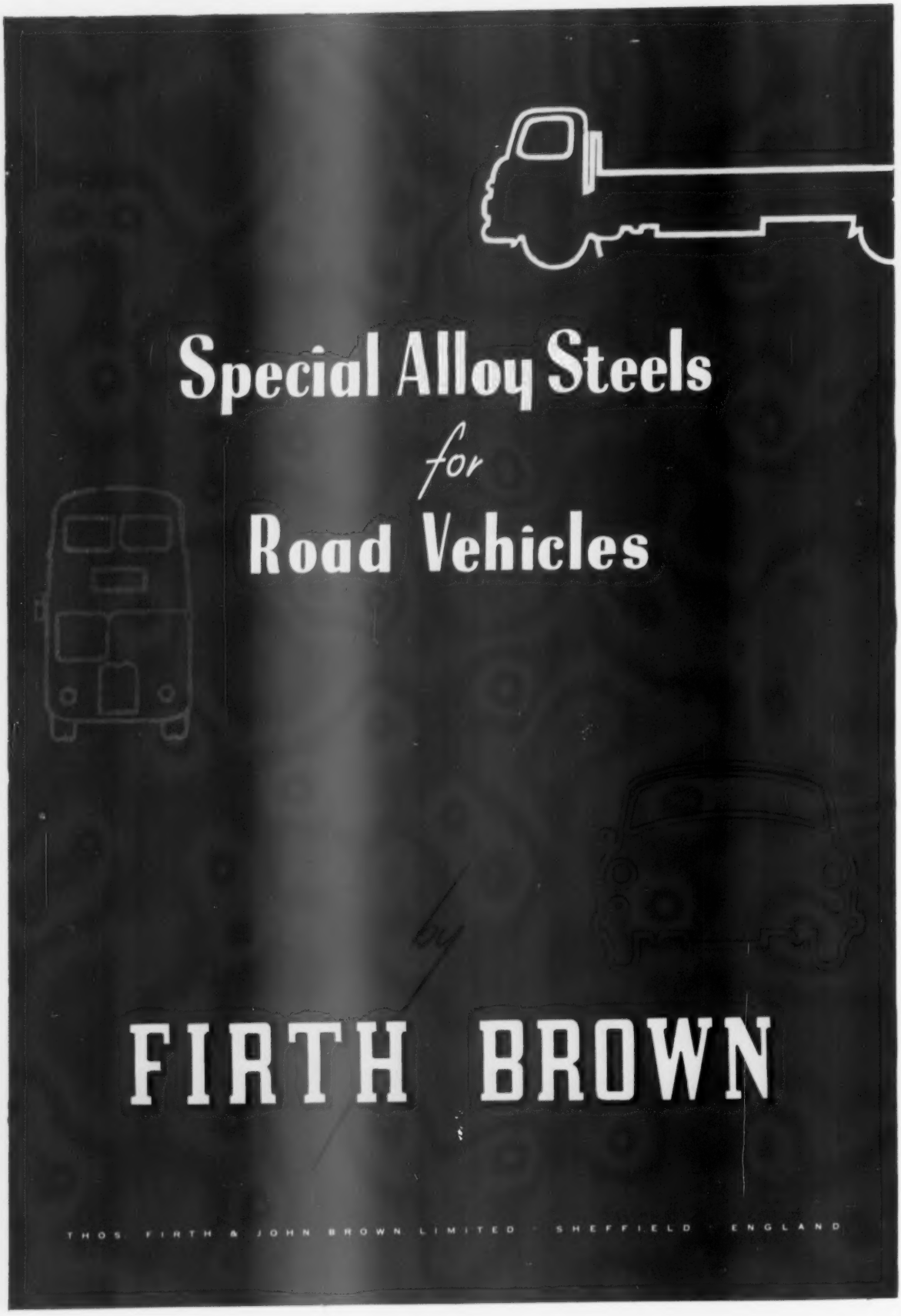
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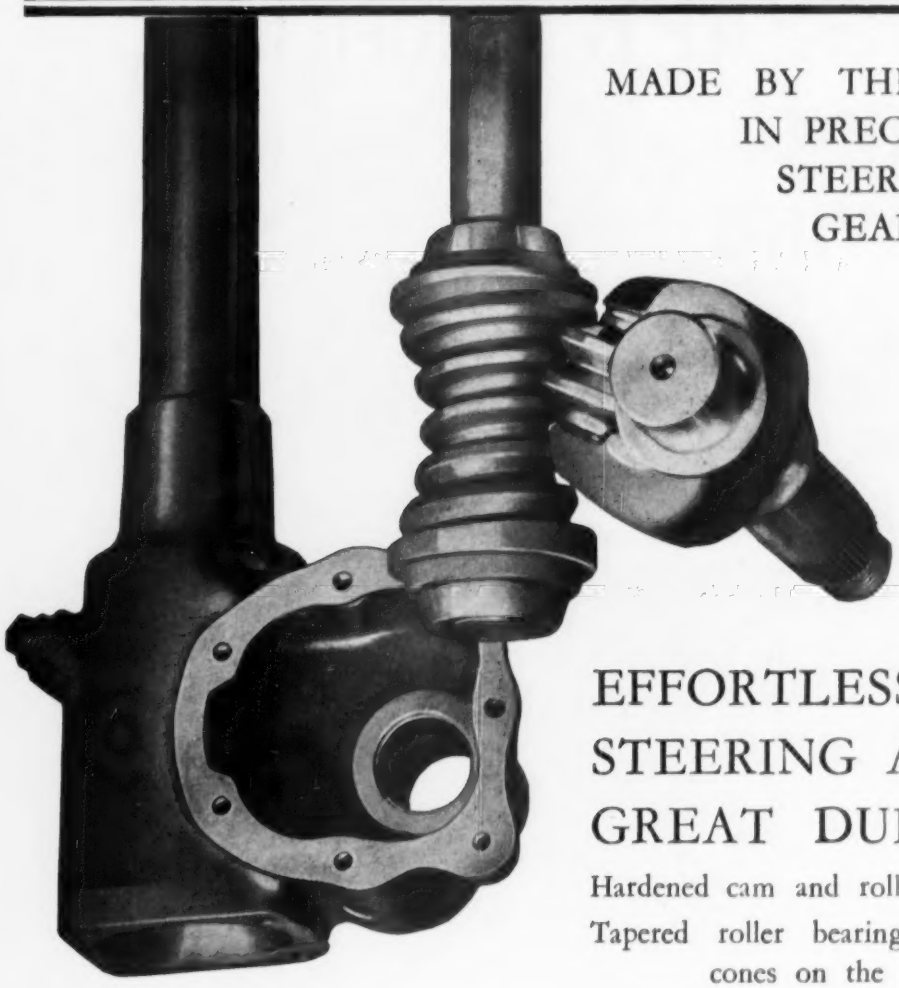

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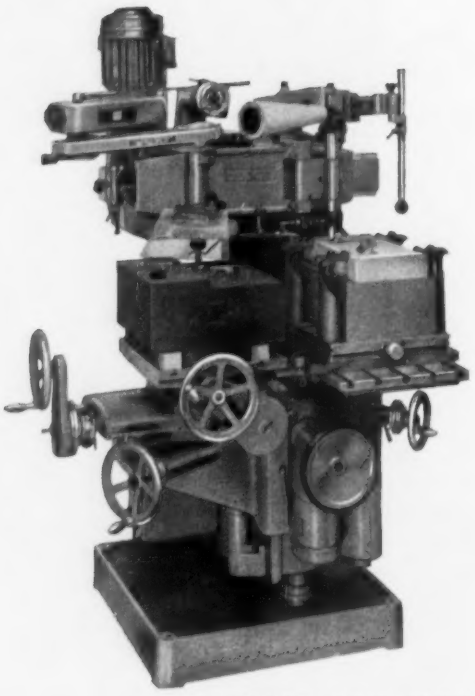

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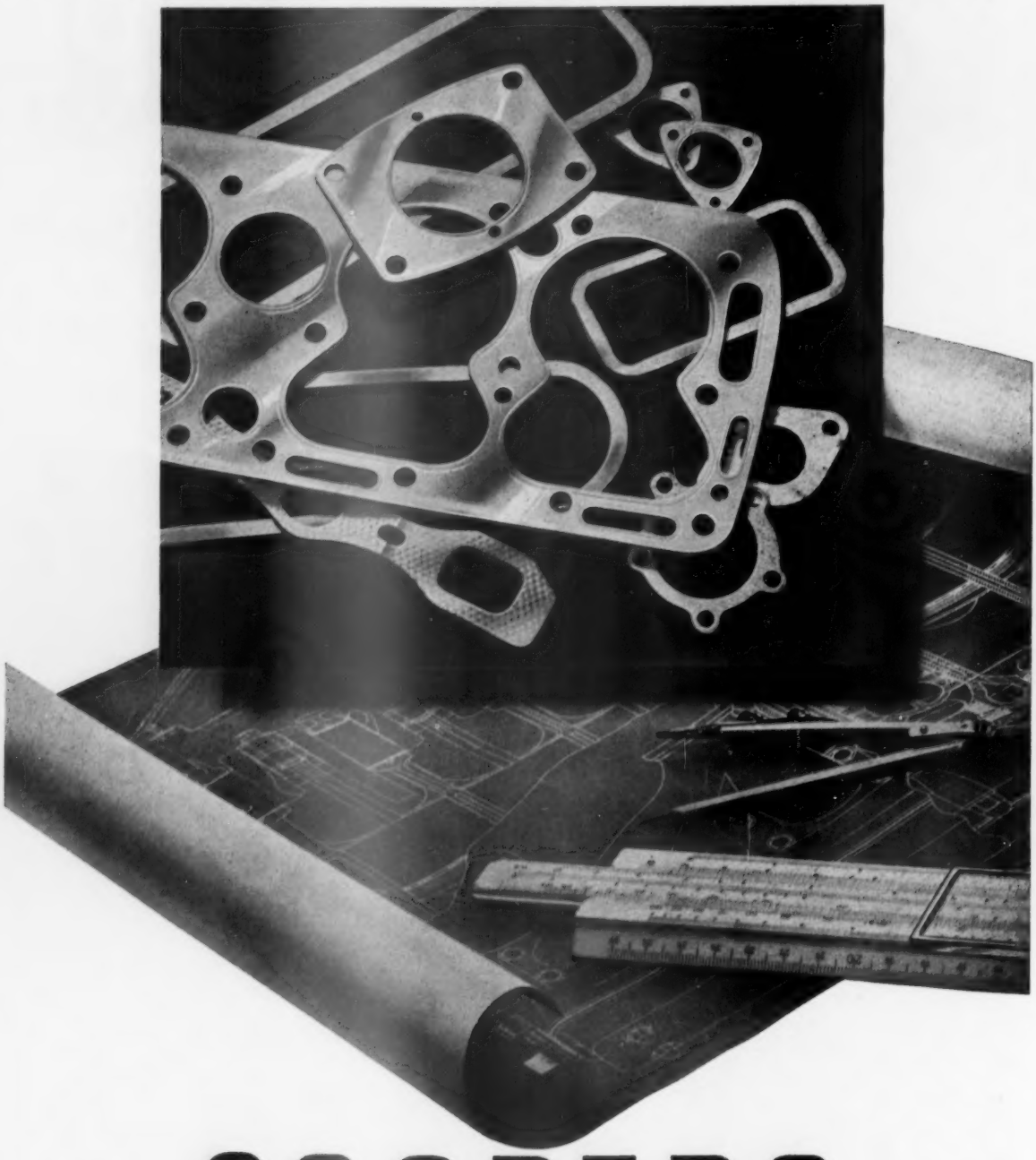

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Coopers' engineers are available for consultation on your sealing problems at all times.

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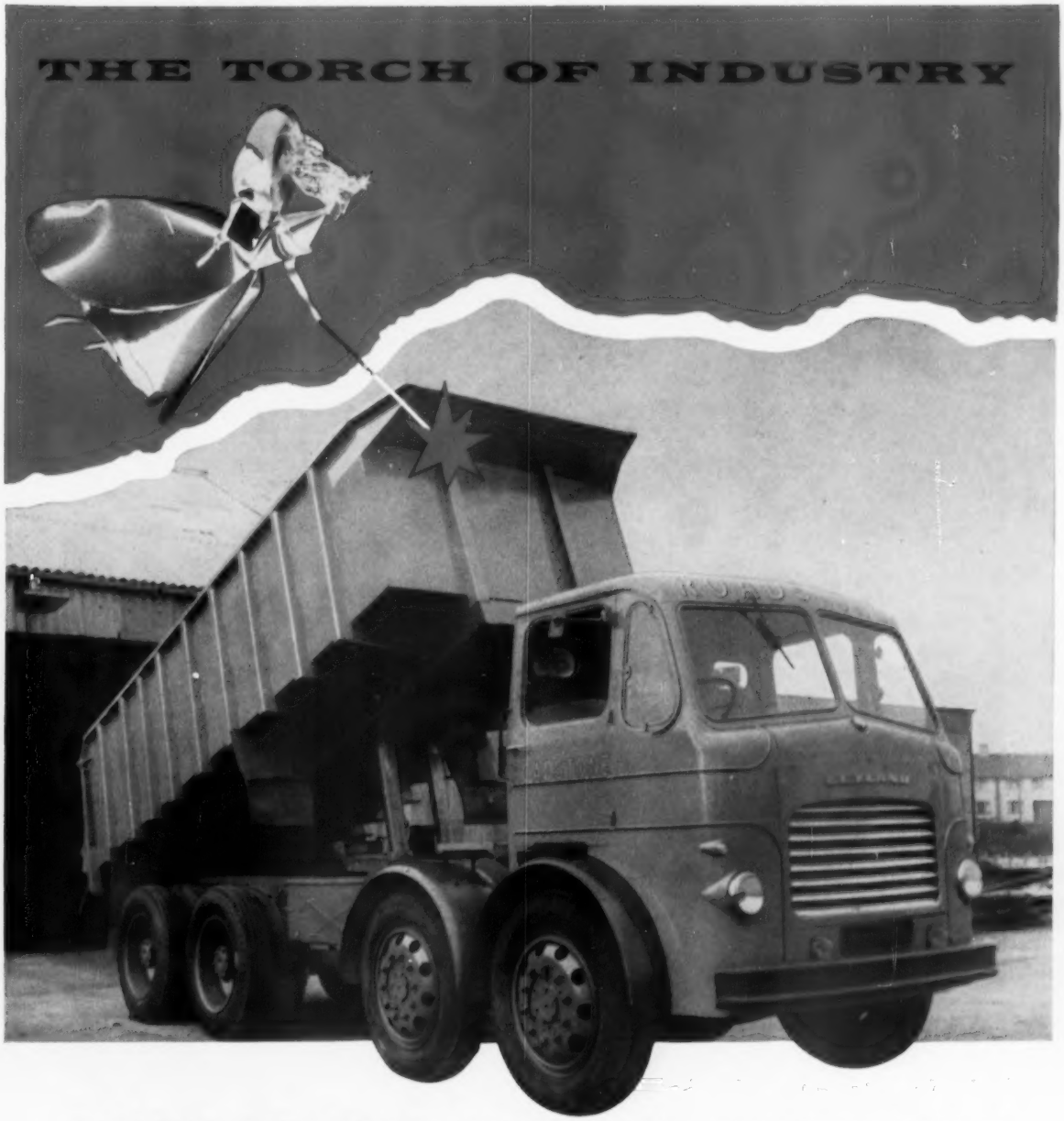


FASTER WELDING WITH ARGONAUT

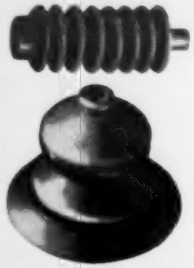
Roadstones Ltd. of Dublin are using the Argonaut welding process to fabricate lightweight, yet sturdy light alloy bodies for their fleet of heavy vehicles engaged in the carriage of gravel, ballast etc. Argonaut has resulted in considerable weight saving over the conventional riveted body, and in increased strength as well. Fabrication time is also reduced. Registered pay load is approximately 16 tons.

BOC BRITISH OXYGEN

Two leading suppliers of engineered rubber under one roof

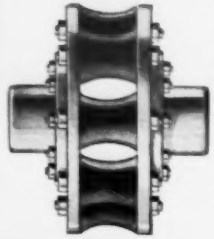
EMPIRE RUBBER COMPANY

RUBBER BONDERS LIMITED



Delicate, complete mouldings to critical dimensions; accurate in every respect; exact in operation. When such problems come to mind call on EMPIRE RUBBER COMPANY. Their increasingly wide variety of rubber components, produced to every sort of industrial specification, do a great deal towards helping machinery run more silently, smoothly and—above all—economically.

These are but two of many available examples of what can be done to bring rubber in as an integral part of a machine.



A metal fixing-piece is bonded to each end of a cylinder of rubber. The cylinder then supports the load in shear or compression—at the flexibility you want. This is a very elementary example of the many productive ways in which rubber-bonded-to-metal components are used. RUBBER BONDERS LIMITED were pioneers in this growing industry. Today they lead both in quality, resourcefulness and variety of products.

EMPIRE RUBBER COMPANY AND RUBBER BONDERS LIMITED DUNSTABLE

R.B. 89

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PISTON RINGS
CYLINDER LINERS
VALVE SEAT INSERTS
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TAPPETS ETC.

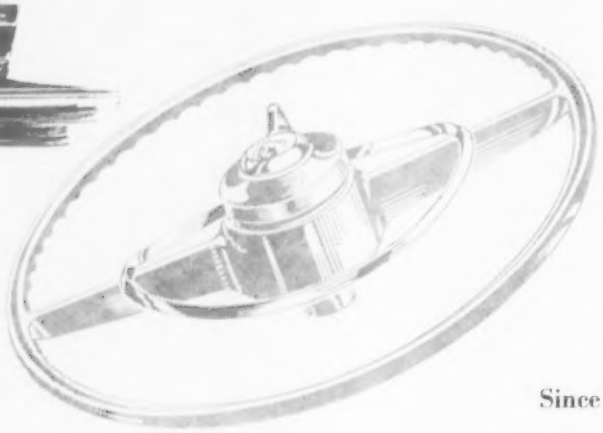
QUALITY PAYS!

THE BRITISH PISTON RING CO. LTD., HOLBROOK LANE, COVENTRY

Cogent

NO ROOM

for complacency here!



Since 1946 we have produced over four million Steering wheels. This gives us no grounds at all for complacency. We are ever striving to improve our technique and our service so as to enhance goodwill and produce always the latest and the best.

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CLIFFORD COVERING *Co Ltd*

A member of the Clifford Group of Companies

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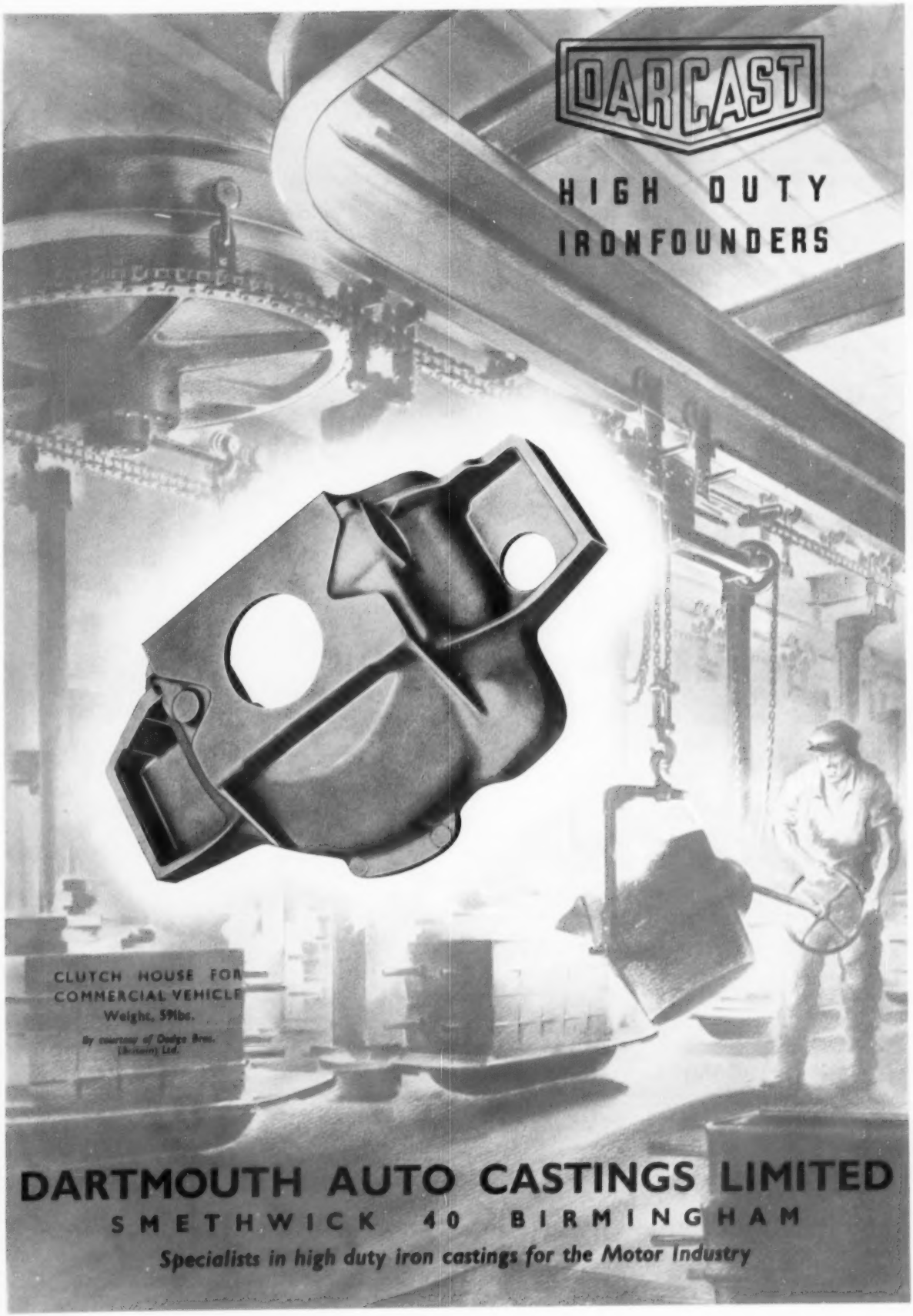
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CLUTCH HOUSE FOR
COMMERCIAL VEHICLE
Weight, 59lbs.
By courtesy of Dodge Bros.
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DARTMOUTH AUTO CASTINGS LIMITED
SMETHWICK 40 BIRMINGHAM
Specialists in high duty iron castings for the Motor Industry

WOODHEADS of LEEDS

The name that means everything in suspensions

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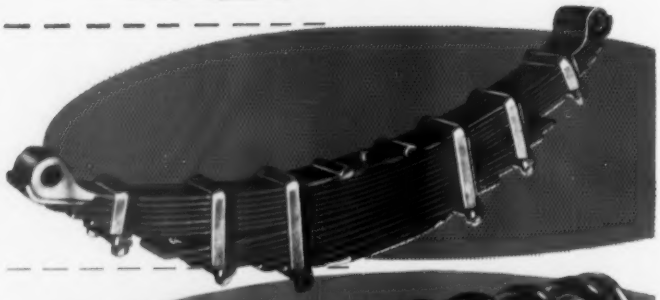
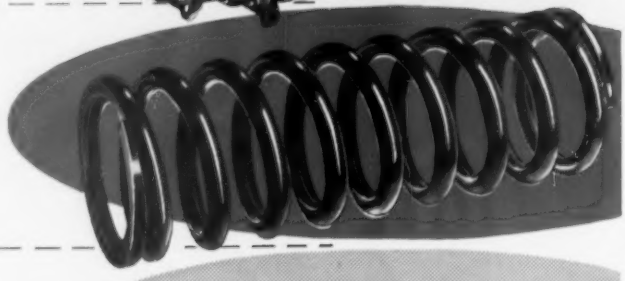
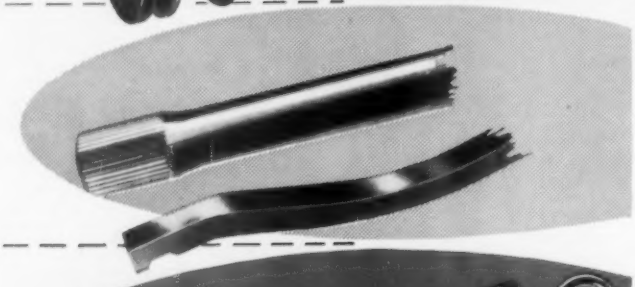
SPRINGS



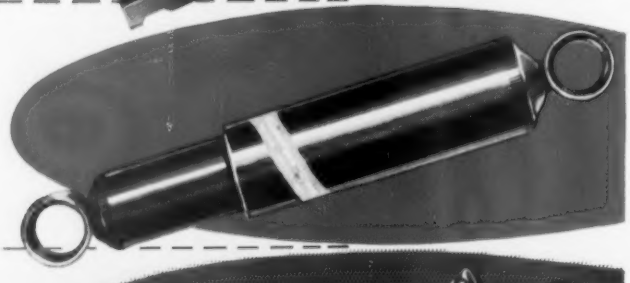
COIL
SPRINGS



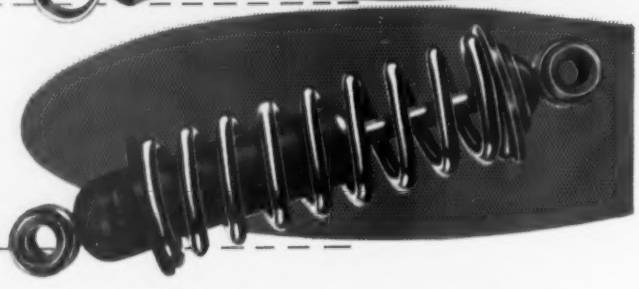
TORSION AND SWAY
BARS



SHOCK
ABSORBERS



SUSPENSION
UNITS



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AND SONS LIMITED
KIRKSTALL ROAD, LEEDS, 4.

Acknowledged to be the largest organisation of its kind in the world. Woodheads of Leeds manufacture every type of suspension equipment for *all* types of road vehicles, including shock absorbers, laminated springs, coil spring and torsion and sway bars.

WOO.4529

50,000 miles more paylife for your fleet vehicles!

Sound like an extravagant claim? It isn't—when you examine the facts about Clayton automatic chassis lubricators.

1 No idle vehicles, no downtime—each vital chassis point is *automatically* lubricated at periodic intervals while the vehicle is on the job, carrying payloads.

2 Because lubrication is *complete* and *constant*, chassis bearing wear is reduced to a minimum, a promise of longer chassis life.

Add to these the many other advantages of a Clayton automatic chassis lubricator—reduced maintenance costs; fewer replacement parts needed; less lubricant used; lower labour costs—and it's easy to see where your 50,000 miles of extra paylife come from.

'Phone or write us for more information. A qualified Clayton engineer will be glad to inspect your fleet and make specific recommendations.

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CLAYTON DEWANDRE CO. LTD.

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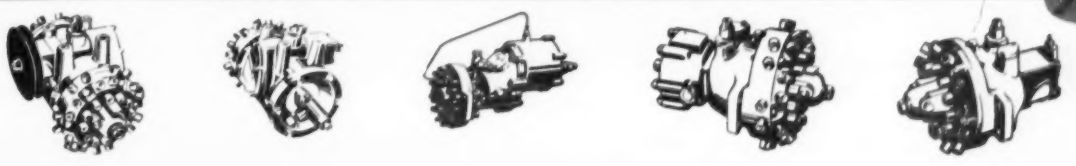

L.20

BRAKING PROBLEMS No 3

Troubled by SHORT DRUM LIFE?

Get the WOVEN-or-MOLDED question sorted out!

For every braking system there is a liner that is just right. It may be molded, it may be woven, and it may be any one of a dozen qualities of either type. At all events, it takes an expert to know precisely which type is needed. That's where the practical experts at the DON depots can help. They *understand* braking—after all, it's their job—and they can supply, from their comprehensive stock of DON woven *and* molded liners, the precise liner to rule out every idiosyncrasy like excessive drum wear in any vehicle.

Make full use of this service.

Fit Don BRAKE LINERS *to be sure!*

The best of Woven and Molded

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GLASGOW C2 *Central 4595*
HULL *Central 52072*
IPSWICH *3023*
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The Greeks had no word for it

Great epics have been written—said the Managing Director—
about the glory that was Greece and so on and so forth but,
for myself, I regard the Ancient Greeks as rather Small Beer.
As every schoolboy knows, they sat outside the City of Troy
for ten solid years arguing and quarrelling before they thought up
the rather childish idea of getting inside
by equipping a Horse with Manpower.

We have changed all that.
Today we equip the Man with Horsepower and
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Desoutter Bros. Limited, The Hyde, Hendon, N.W.9. Telephone: Colindale 6346 (5 lines). Telegrams: Despnuco, Hyde, London.
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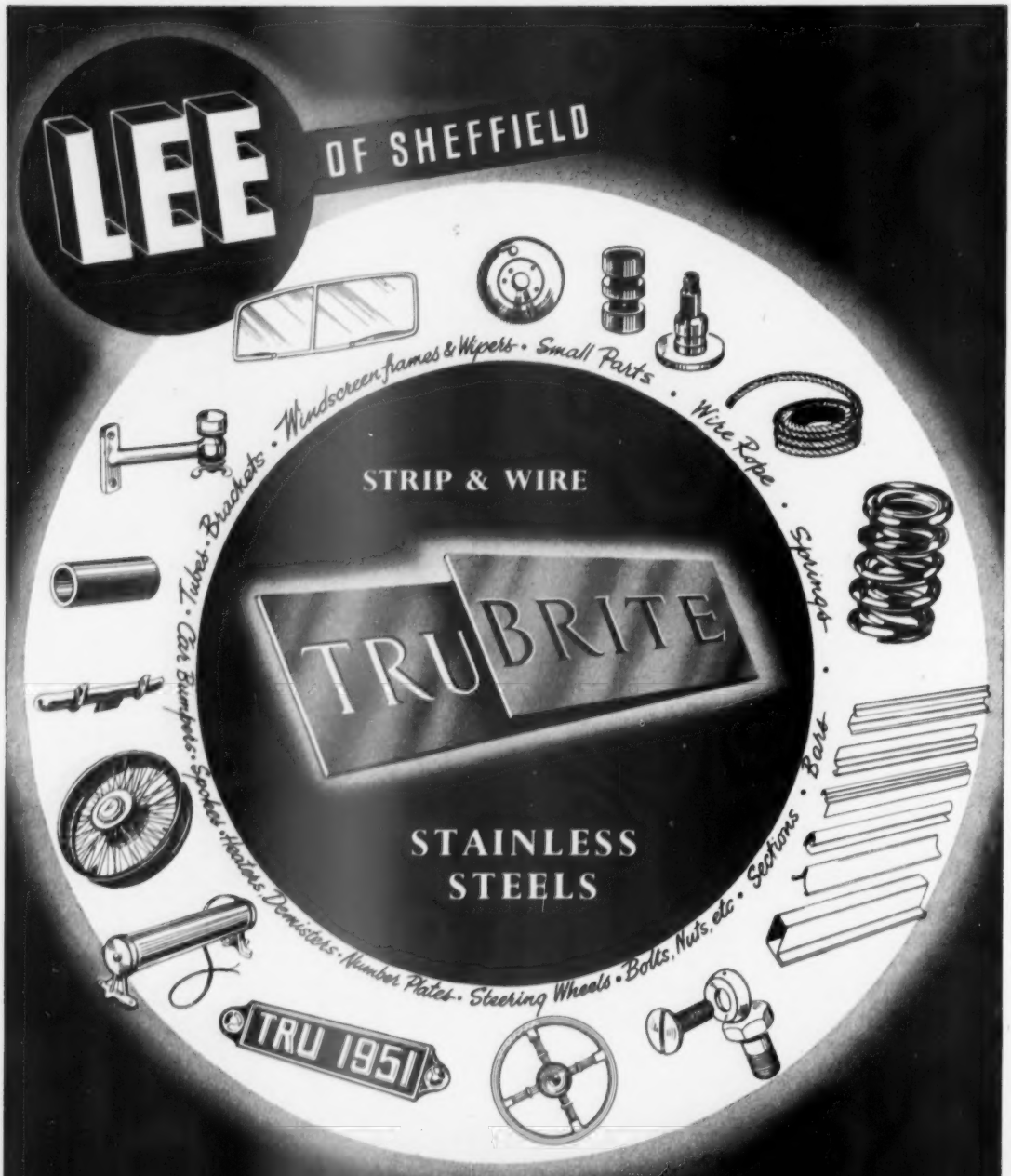
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The Motor Car Industry is another example of the ever increasing uses of "Trubrite" Stainless Steel in the form of Cold Rolled Strip and Wire, quite apart from the already well established uses of Bright Mild Carbon Steel Strip, Bright Steel Bars and High Strain Wires, with which we have for so long retained the confidence of Motor Car and Motor Accessory makers.

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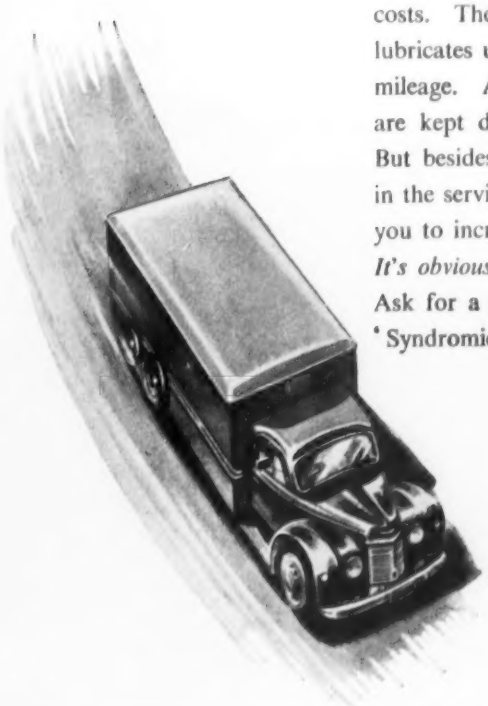
C890-2

*'Syndromic' Automatic Lubrication soon pays for itself!

*INCREASES YOUR PAYLOADS 10%



From the moment you equip your vehicle with 'Syndromic' Lubrication it begins to pay for itself! First, it saves up to 75% on lubrication costs. Then, it ensures less wear on bearings because it automatically lubricates up to 80 bearings (using 2 pumps) at periods controlled by mileage. As a result, maintenance and replacement time and costs are kept down to a minimum.

But besides all that, 'Syndromic' lubrication saves the hours spent in the service bay. *It lubricates while the vehicle is working*, allowing you to increase your vehicles' payloads by as much as 10%!

It's obvious that 'Syndromic' lubrication soon pays for itself!

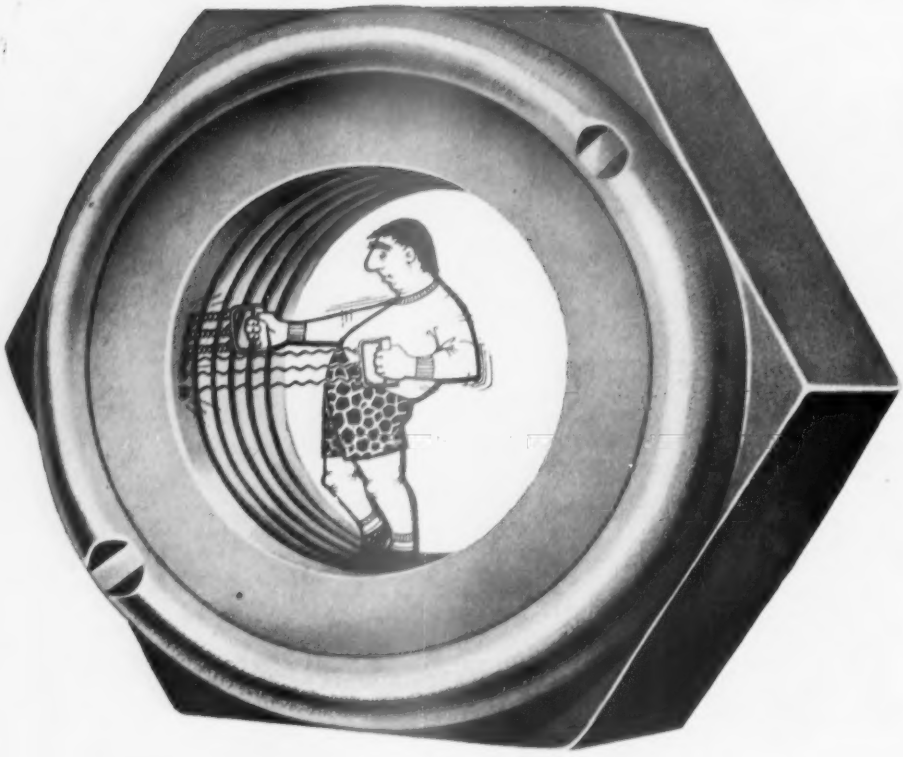
Ask for a Tecalemit technical representative to call and discuss the 'Syndromic' system with you.

TECALEMIT

The Authority on Lubrication

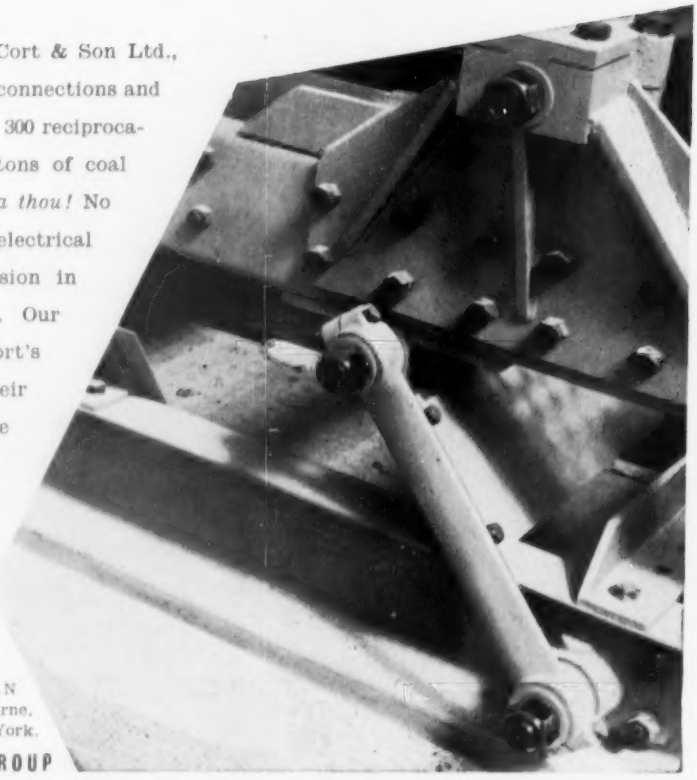
PLYMOUTH, ENGLAND

T417



. . . but nothing moves the NYLOC

On Heavy Duty Screens made by Robert Cort & Son Ltd., Reading, Nyloc nuts secure the driving rod connections and the cross shafts. These screens make about 300 reciprocations per minute and handle up to 300 tons of coal per hour. *Yet the Nyloc nuts never budge a thou!* No amount of recurring tension (vibration, electrical impulses, temperature changes or immersion in oily liquids) can relax their hold on the bolt. Our illustration shows Nyloc nuts fitted on Cort's light duty screen. Equally tenacious in their own particular way are Fibre Nuts (with fibre insert) and Pinnacle Nuts (with metal diaphragm). We'll be glad to advise you which type of self-locking nut is best for your job.



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Buying a brake drum is a grave responsibility. Human lives and the safe delivery of merchandise are dependent upon the capacity of the brake drum to resist enormous heat without damage to the braking surface. The Meehanite Research Institute has conducted intensive study into this problem over a period of 30 years. You buy the results of this 30 years of research in every Meehanite brake drum.

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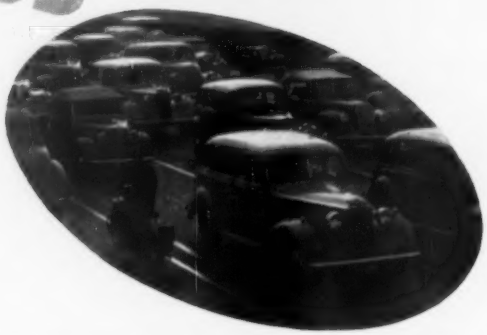
FITTED FOR GOOD



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Consummate

Quality

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Workmanship

The delivery valves of a fuel injection pump require most careful design and extreme accuracy

in manufacture. Acting as high pressure non-return valves, they also provide for instantaneous relief of pressure in the fuel line, so necessary for efficient engine operation. The parts of the valve are finished by hand lapping.

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MARK 4 **MINIATURE CONNECTORS**

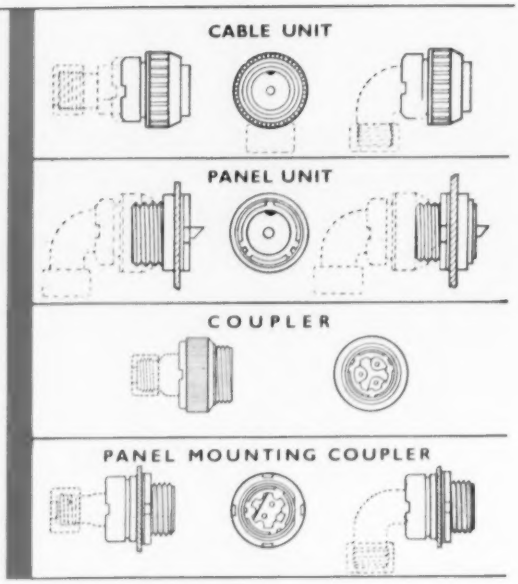
. . . the most flexible system of interconnection yet devised

Compact, miniature Plugs and Sockets of Plessey manufacture enabling multiple connections of great complexity to be made rapidly and without fear of error, have revolutionised the design of electrical and electronic equipment of all types.

Furthermore, they make possible prefabrication of complete wiring systems on the factory bench, with a consequent saving in production time.

Rapid disconnection for servicing of equipment units, high electrical and mechanical efficiency and the absolute flexibility of the system are other outstanding advantages.

The Mark 4 range is available both in aluminium (single hole fixing) and brass (single hole *or* flange fixing).



These are the advantages . . .

Weather and Pressure Proof • Adaptable, Foolproof Unit Connection • Reliable Low-Resistance Contact • Quick Disengagement for Routine Testing • Allow Considerable Prefabrication of Wiring Forms • Numerous Pin Combinations • Robust and Durable • Eliminate Risk of Connection Errors after Servicing •

The Mk4 range of **Plessey** Plugs and Sockets is described fully in PUBLICATION No. 674 . . . See that your Design Engineers have a copy

THE PLESSEY COMPANY LIMITED · ILFORD · ESSEX

PBI

HORIZONTAL

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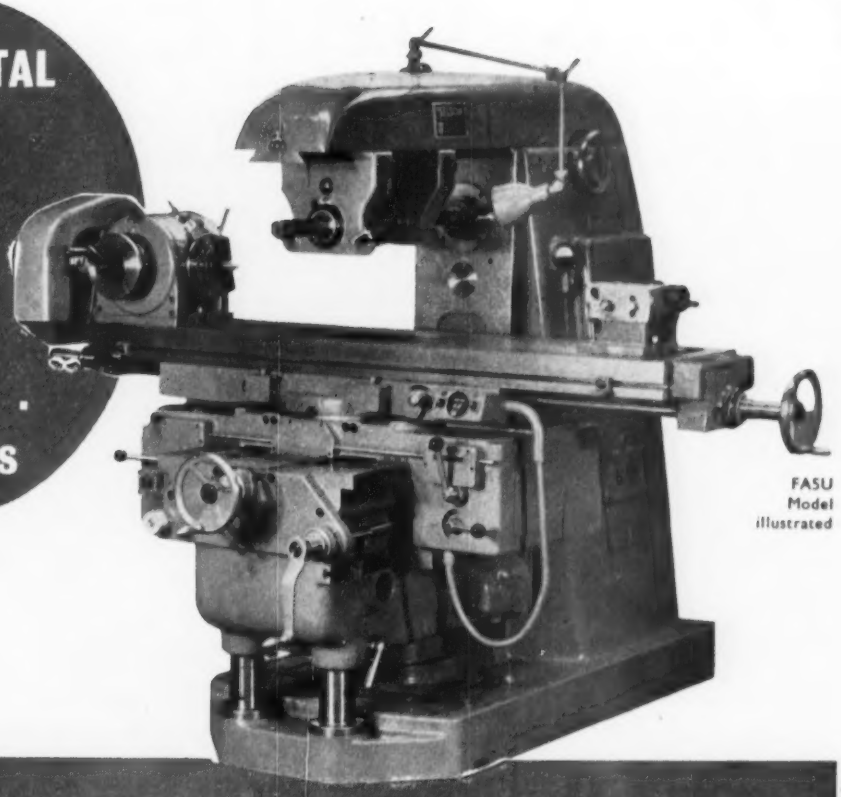
All sizes . . .

. . all types



FASU Model illustrated

ZBROJOVKA MILLERS

Early delivery of all models from London stock

BRIEF SPECIFICATIONS				
FOUR SIZES	No. 2	No. 3	No. 4	No. 5
Working surface of table (approx.)	40" x 8"	50" x 10"	64" x 12"	80" x 16"
Power longitudinal travel (approx.)	25"	32"	40"	50"
Power cross travel (approx.)	8"	8"	12"	16"
Spindle Speeds	63-2800	45-2000	32-1400	16-1400

SOLE AGENT

Selson MACHINE TOOLS

The Selson Machine Tool Co. Ltd

CUNARD WORKS, CHASE ROAD, NORTH ACTON, LONDON, N.W.10

Telephone: Elgar 4000 (10 lines) Telegrams: Selsomachi, London

THE 600 GROUP

PARK GATE

QUALITY STEELS FOR DROP STAMPINGS

black bars
for
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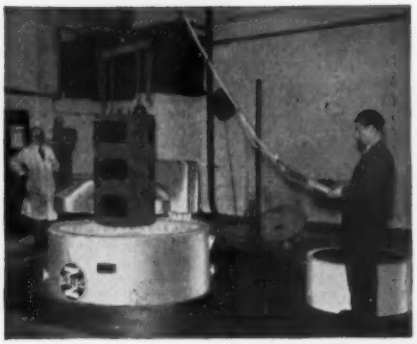
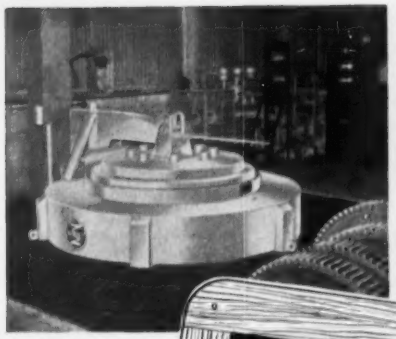
PARK GATE IRON & STEEL CO. LTD., ROTHERHAM
TELEPHONE: ROTHERHAM 2141 (10 lines) TELEGRAMS: YORKSHIRE, PARK GATE, YORKS



GAS CARBURISING

... that's correct



top: two 100 kW units, 27" diam. x 54", for heavy gear parts.

centre left: gas fired unit, 48" diameter x 48" deep, using endothermic atmosphere generator.

centre right: the smallest standard pit unit; 40 kW furnace with drip-feed carburising system.

bottom: two 65 kW furnaces, 24" diameter x 42" deep, with (rear) pit tempering furnace and (front) the gas atmosphere generator.

Birlec Publication No. 70 gives full details; it will gladly be sent.

- ✓ Gas Carburising is now accepted as the most economical - and most efficient - means of imparting, controlled hardened case depths to engine, transmission, machine tool and other components.
- ✓ Birlec Limited has over 70 standard pit gas carburising furnaces in commission, providing comprehensive experience on atmosphere control, loading and charge handling techniques and time/temperature cycles.
- ✓ Both electric and gas-heated designs are available.
- ✓ Atmosphere production is either by enriching carrier gas, provided by an endothermic generator, or by the liquid "drip-feed" process.
- ✓ Standard charge space dimensions vary upwards from 18" diameter X 30" deep to 48" diameter × 48" deep.

BIRLEC

BIRLEC LIMITED

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Also LONDON · SHEFFIELD · GLASGOW · NEWCASTLE-ON-TYNE

SM/B2462/55

Simply designed
FOR THE DRAUGHTSMAN

Down a little! *Still nearer!* *Right on the job!*

How draughtsmen fared before ANGLEPOISE revolutionized drawing board lighting we simply don't know! This fine lamp is just indispensable for fine accurate work. It throws its clear shadowless concentrated light right on the work from any angle at a finger touch and 'stays put' until a new position is called for. It can be mounted separated from the board or on an adjustable swivel base for sloping boards.

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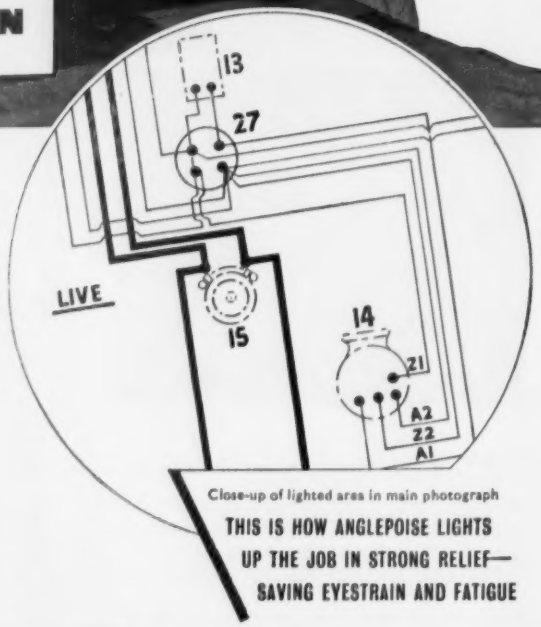
Sole Makers: HERBERT TERRY & SONS LIMITED
REDDITCH · WORCESTERSHIRE · ENGLAND



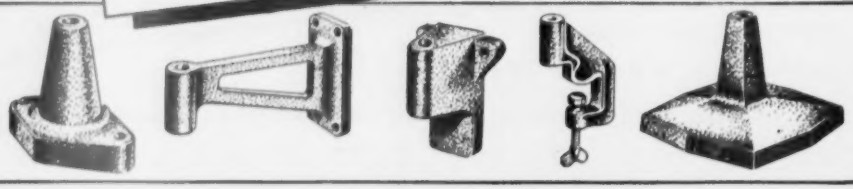
Close-up of lighted area in main photograph

THIS IS HOW ANGLEPOISE LIGHTS UP THE JOB IN STRONG RELIEF— SAVING EYESTRAIN AND FATIGUE

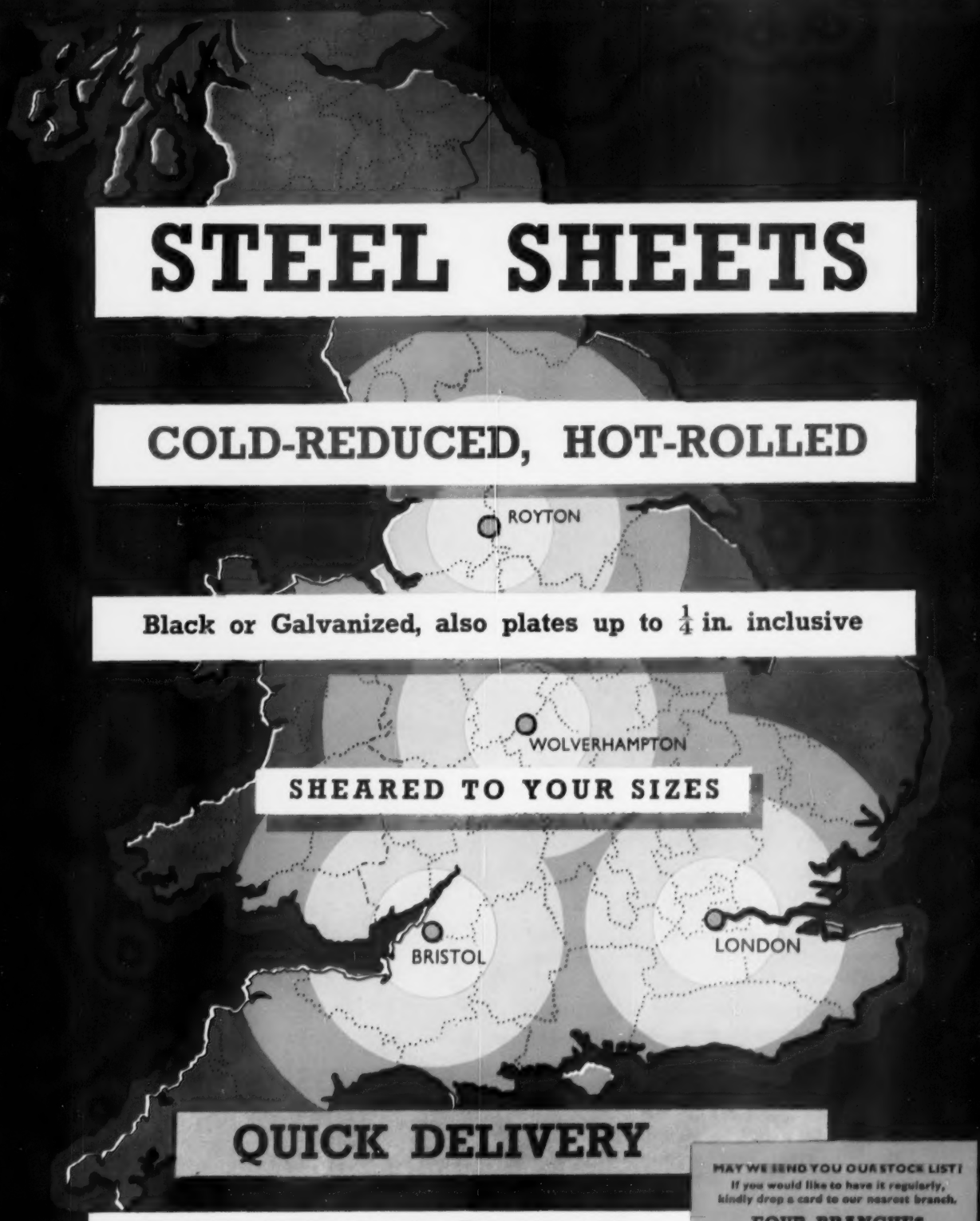
TERRY ANGLEPOISE LAMPS

Pat. all countries

SOME ALTERNATIVE BASES FOR ALL MODELS



HT18



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Black or Galvanized, also plates up to $\frac{1}{4}$ in. inclusive

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TIME
FIVE
MINUTES

SOLUTION....

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A MAGNIFICENT SALES AID FOR YOUR PRODUCTS

Pull Tear-Tab on Container wall and "zip" it's open... No damage to contents through knives or other tools

A BOON FOR "SELF-SERVICE" DEPARTMENTS CHEAPENS HANDLING AND LABOUR COSTS AND SPEEDS UP RE-STOCKING OF COUNTERS

Ask your usual Container Manufacturer for "RIPPATAPED" Cases. In case of difficulty—contact us for list of Manufacturers Rippatape Equipped. Help <u>your</u> products sell faster and build goodwill with your customers

SEND FOR ILLUSTRATED FOLDER OR
ASK OUR REPRESENTATIVE TO CALL

TIME
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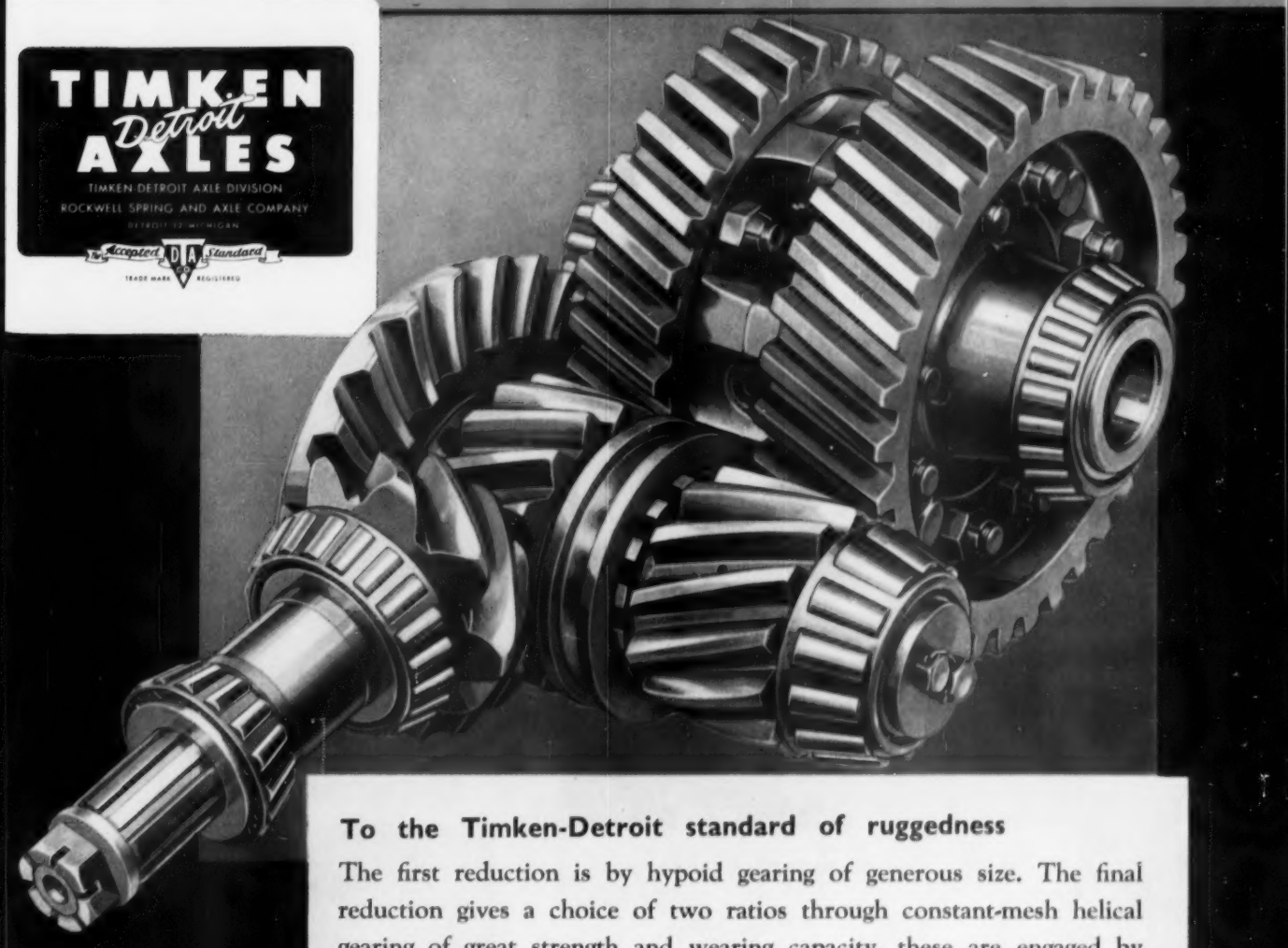
TIMKEN-DETROIT AXLES

With 2-Speed Hypoid and Helical Drive

TIMKEN Detroit AXLES
TIMKEN-DETROIT AXLE DIVISION
ROCKWELL SPRING AND AXLE COMPANY
DETROIT 32, MICHIGAN
The Accepted D A Standard
TRADE MARK REGISTERED



To the Timken-Detroit standard of ruggedness

The first reduction is by hypoid gearing of generous size. The final reduction gives a choice of two ratios through constant-mesh helical gearing of great strength and wearing capacity, these are engaged by robust dog-clutches.

High-duty Timken tapered-roller bearings: high-duty differential.

Brakes: hydraulic-dual primary or air-actuated.

Front axles are also available.

Exclusive European distributors for The Timken-Detroit Axle Division, Rockwell Spring and Axle Company, Detroit, Michigan, U.S.A.:—

AUTOMOTIVE PRODUCTS COMPANY LIMITED
BROCK HOUSE, LANGHAM STREET,
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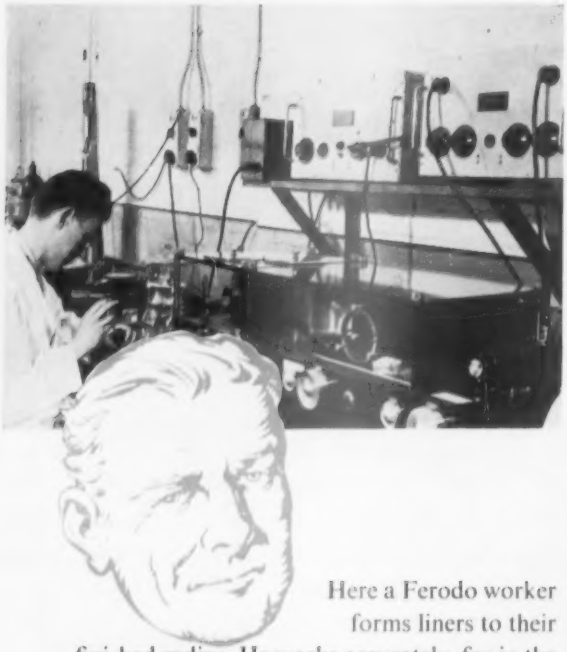
GUARANTEED AXLE-SHAFTS AND AXLE HOUSING

Timken-Detroit torsion-flow axle-shafts, heat-treated and shot-peened, are guaranteed for 100,000 miles of operation or for three years, whichever occurs first, against defective materials and workmanship. The forged axle housing is guaranteed for the life of the vehicle.

THREE MEN O

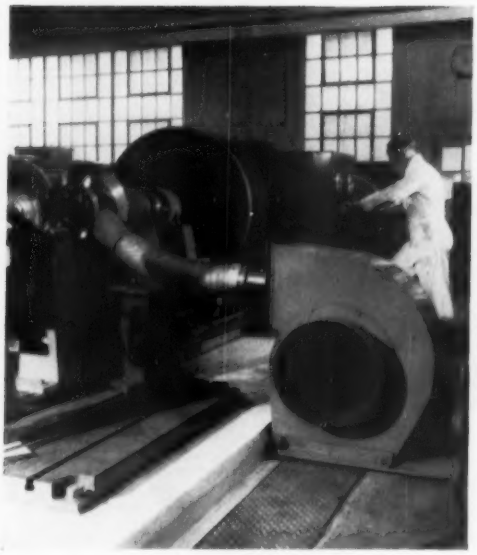
These three men are doing completely different jobs,
yet they have a common purpose . . . they are all making sure that
Ferodo Anti-Fade Brake Linings are safe, hard wearing,
with the highest possible resistance to fade.



The physicist in the laboratory investigates basic materials. Only a full understanding of their properties can ensure the accurate composition of every type of Ferodo Lining. Some of Ferodo's research is completely original and helps to increase scientific knowledge generally.

Here a Ferodo worker forms liners to their finished radius. He works accurately, for in the manufacture of Ferodo Linings the emphasis is on precision. Whether it is a hydraulic press or a giant multiple drill, every machine is in itself the result of research and experience. This fact, combined with strict examination at every production stage, ensures that on any make of car in the world the correct Ferodo Lining will always fit perfectly.

FERODO . . .



A Ferodo technician reads off results as a brake lining undergoes a test far harsher than normal road usage. In the uniquely equipped Test House, experimental linings are tried and proved, as well as linings selected at random from normal production batches. Further tests are made by the powerful cars of the Ferodo Test Fleet both on the road and on the M.I.R.A. track.

• • •

FERODO FACILITIES FOR RESEARCH AND TESTING ARE FREELY AVAILABLE TO THE MOTOR INDUSTRY AT ALL TIMES

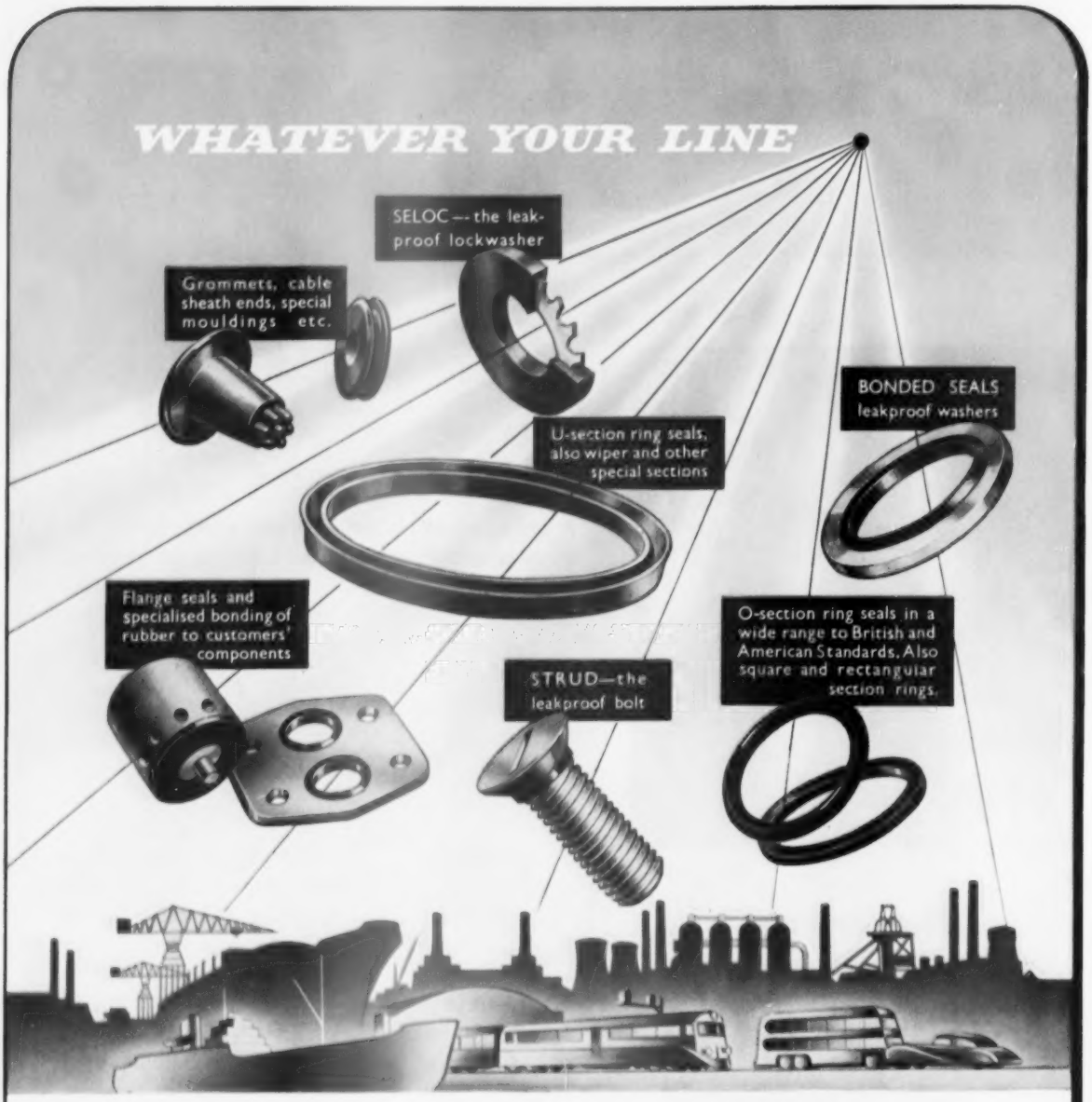
The men of Ferodo have the most up-to-date equipment and machines with which to work. To these they bring precious skill, experience and pride in the finished product.

The result is shown in the world-wide reputation of Ferodo Anti-Fade Brake Linings . . . their long record of successes in all types of international speed events . . . and the confidence placed in them by British car manufacturers.

FERODO

***ANTI-FADE* BRAKE LININGS**
CLUTCH FACINGS

FERODO LIMITED • CHAPEL-EN-LE-FRITH *A Member of the Turner & Newall Organisation*


WHATEVER YOUR LINE
SELOC—the leak-proof lockwasher
Grommets, cable sheath ends, special mouldings etc.
BONDED SEALS leakproof washers
U-section ring seals, also wiper and other special sections
Flange seals and specialised bonding of rubber to customers' components
O-section ring seals in a wide range to British and American Standards. Also square and rectangular section rings.
STRUD—the leakproof bolt

Whether it be an oil, grease, gas, alkali, dilute acid or any other fluid that must be sealed, whether pressures are great or small, or the temperature high or below zero, there is a Dowty Seal to suit your specific requirements.

Our extensive design and research departments are always ready to advise in the selection of the most suitable specification and media for your own particular application.

DOWTY Seals

Member of the DOWTY Group

DOWTY SEALS LIMITED · ASHCHURCH GLOS.

Telephone Tewkesbury 2271

Using stock bushes means more productive plants

We all know how a hold-up for some particular part can upset the production flow. This need not happen with bronze bearings and bushes. Our range of stock sizes offers such a wide selection based on variation in inside and outside diameters combined with graduated lengths, that will completely satisfy the requirements of many of the specially designed bushes. The fundamental difference is that stock bearings really *are* in stock, ready for off-the-shelf delivery. No waiting time; no special making; no special dies; no heavy stocks to carry. All that may be necessary is a small adjustment in mechanical design. We know what our bearings will do; you know what you require of them. Isn't it time we got together? May we send you our Standard Sizes list as a first step?

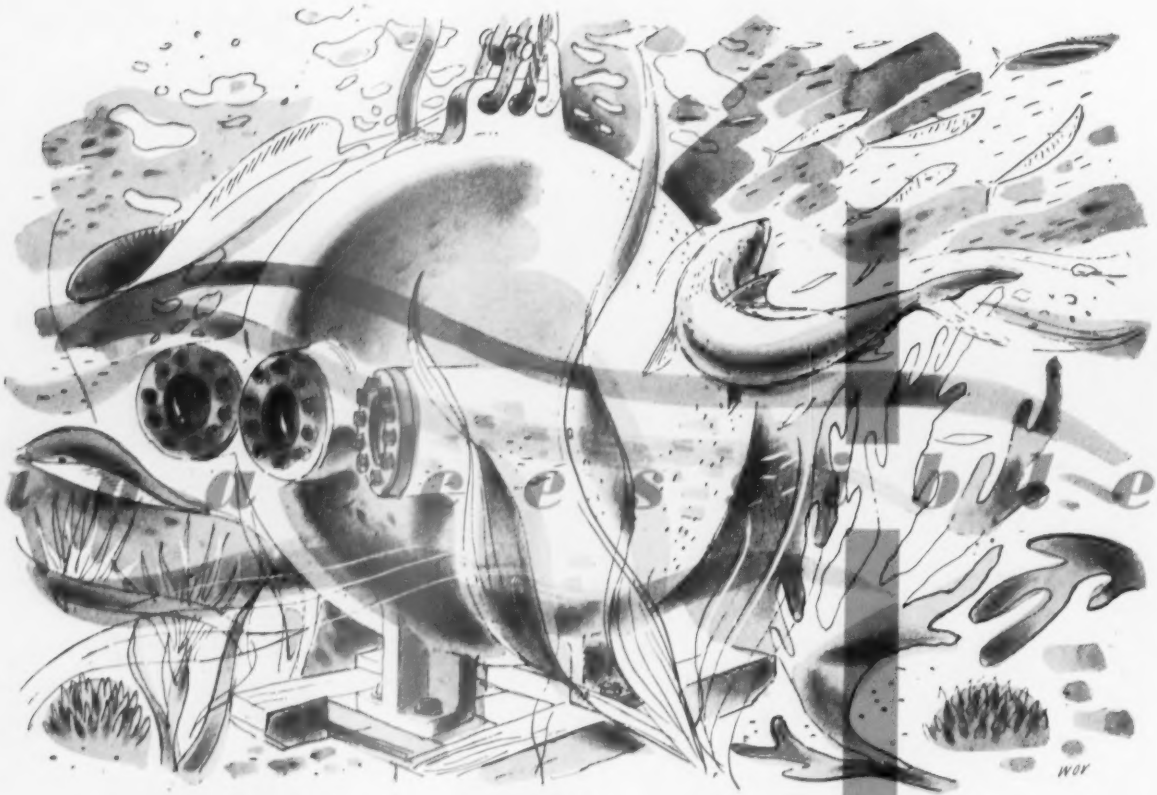
COMPO

Standard Sizes

BRONZE BEARINGS

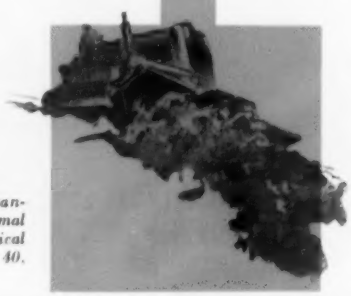
BOUND BROOK BEARINGS LTD., TRENT VALLEY TRADING ESTATE, LICHFIELD, STAFFS

Telephone: Lichfield 2027-2028 *(A Birfield Company)* *Telegrams: Boundless, Lichfield*



Ocean depths *hold the secrets of hitherto untapped resources; in the metabolism of ocean creatures or the geological structures of the ocean bed scientists may find ways of replenishing the decreasing material sources of the above-water world. First, however, must come exploration by diving bell or bathysphere and then the problem of inaccessibility must be solved. In normal industrial and domestic life, too, this problem is present. The automatic lubrication of inaccessible parts is one example — but here "Reservoil" sintered-metal oil-retaining bearings provide the answer. With "Reservoil" bearings oil is constantly fed to the surface by capillary action, ensuring trouble-free running with the minimum of maintenance.*

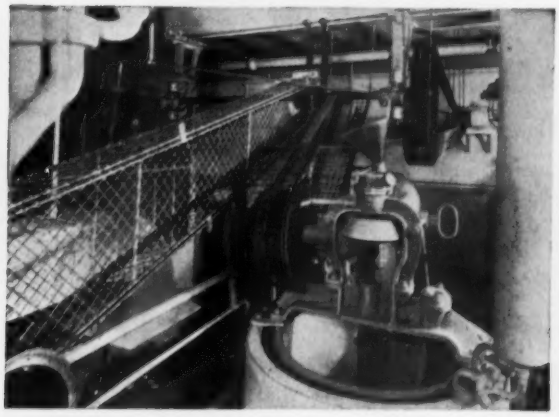
Reservoil
SINTERED METAL
OIL RETAINING BEARINGS



Bulldozers are an instance where "Reservoil" bearings are used to advantage. These bearings are supplied in shapes and sizes for every normal medium duty, but attention is drawn to the range of plain cylindrical bearings to B.S.I. Standard dimensions. Write for Catalogue SD 40.

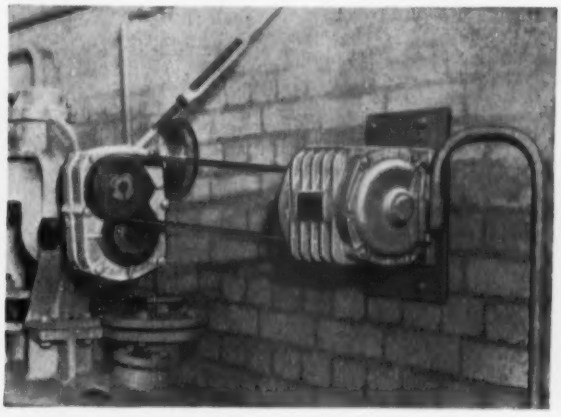
THE MORGAN CRUCIBLE COMPANY LTD., BATTERSEA CHURCH ROAD, LONDON, S.W.11 BATTERSEA 8822
SM 116

NOW! SPACE IS NO PROBLEM IN SPEED REDUCTION



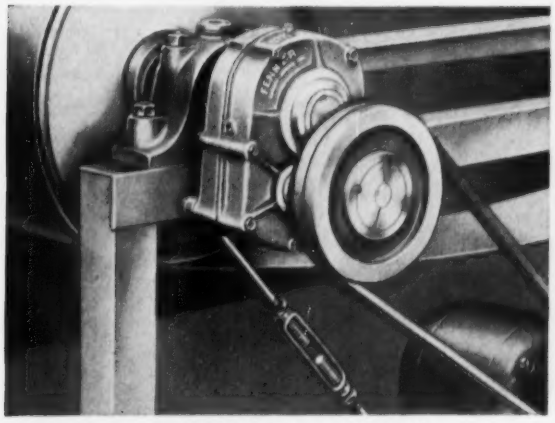
THE OLD METHOD.

A nationally famous firm of manufacturing chemists had the problem of driving chemical mixers at 60 r.p.m. A base mounted gear box was impractical and a cumbersome and expensive double reduction overhead drive had to be used.



THE NEW METHOD.

A Torque-Arm shaft mounted speed reducer was installed on top of the mixer driven through a Taper-Lock V-Drive by a wall mounted motor. Advantages: clean safer drive, economy in space, simplified maintenance, more useful power transmitted.



TORQUE-ARM CAN HELP YOU TOO!—

Space limitations no longer prevent you installing a speed reducer. The Fenner Torque-Arm Reducer mounts direct on the driven shaft and can be driven through any V-Belt Drive. It delivers up to 97% of power to output shaft. No motor base or slide rails are required, no flexible couplings needed. It is suitable for the majority of slow speed drives.

TECHNICAL LITERATURE IS AVAILABLE.

Literature giving all the facts about Fenner Torque-Arm Speed Reducers is available . . . send for your copy TODAY. Fenner Technical Advisory Service will be glad to consider your Speed Reduction problems and advise on the suitable application of Torque-Arm Speed Reducers.

J. H. FENNER & CO. LTD. · HULL · ENGLAND

FENNER

OF HULL

FIRST WITH TORQUE-ARM SHAFT SHAFT MOUNTED *MOUNTED SPEED REDUCERS IN THE U.K.*

In Arctic Cold

or

Desert Heat

VOKES

Pioneers of scientific filtration.

filters give maximum protection

When operating temperatures are abnormally low, it is the first few minutes that count—and because of the intrinsic efficiency and large surface area of the Vokes felt fabric element, the direct-flow device—which must unfailingly operate when the oil is highly viscous—remains open for the absolute *minimum* of time, ensuring that unfiltered oil is not allowed to circulate over dangerously long periods. And at the other end of the temperature scale, Vokes filters give the same unfailing protection . . . that is why they were chosen to protect equipment on all the major Middle East Pipe Lines.

VOKES LIMITED, GUILDFORD, SURREY London Office: 123 Victoria Street, Westminster, S.W.1.

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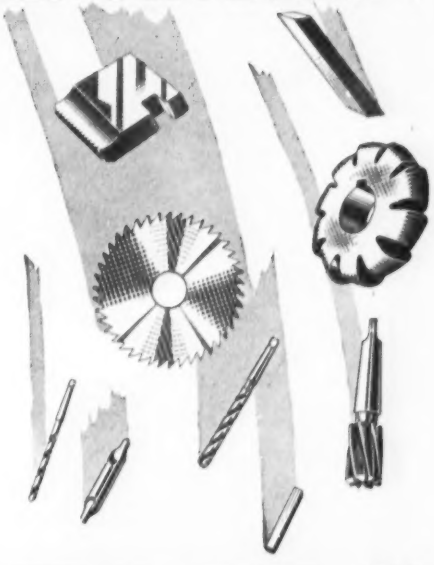
A New SERVICE THAT SOLVES TOOL SUPPLY PROBLEM !



Have you still to experience the advantages of obtaining practically all your requirements in Engineers' Small Tools and Sundries from Hallmac?

This recently formed Organisation has set a new standard in efficiency, with on-time delivery from a large and comprehensive stock range comprised of only the finest tools.

All four Hallmac Depots carry stocks and are strategically placed. Hallmac Representatives are linked throughout the country—Hallmac is ready and waiting to serve you.

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LIMITED

Head Office and Depot: GPO Box No. 39, Macrome Road, Wolverhampton. *Telephone 52001 (5 lines).*
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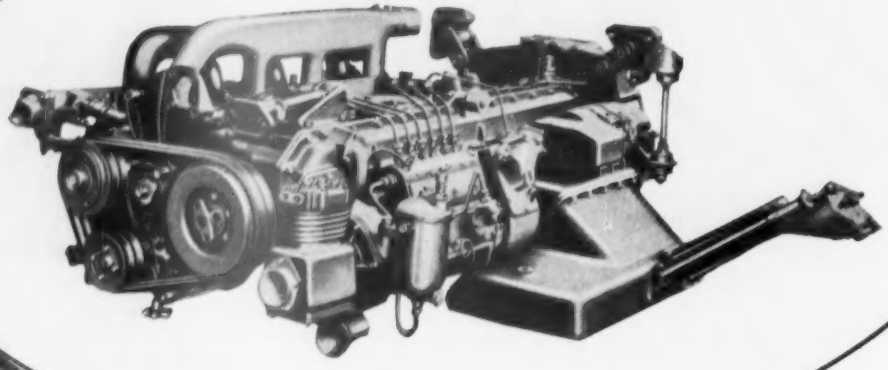
DISTRIBUTORS OF ENGINEERS' SMALL TOOLS
Macrome Treated Tools · Hand and Power Tools

JW.AA 3611

The E.O. 600/95 Flat Engine
in the
LEYLAND ROYAL TIGER CHASSIS

Leyland



LTD

THIS MARK
ON ALL VANDERVELL
PRODUCTS

fit

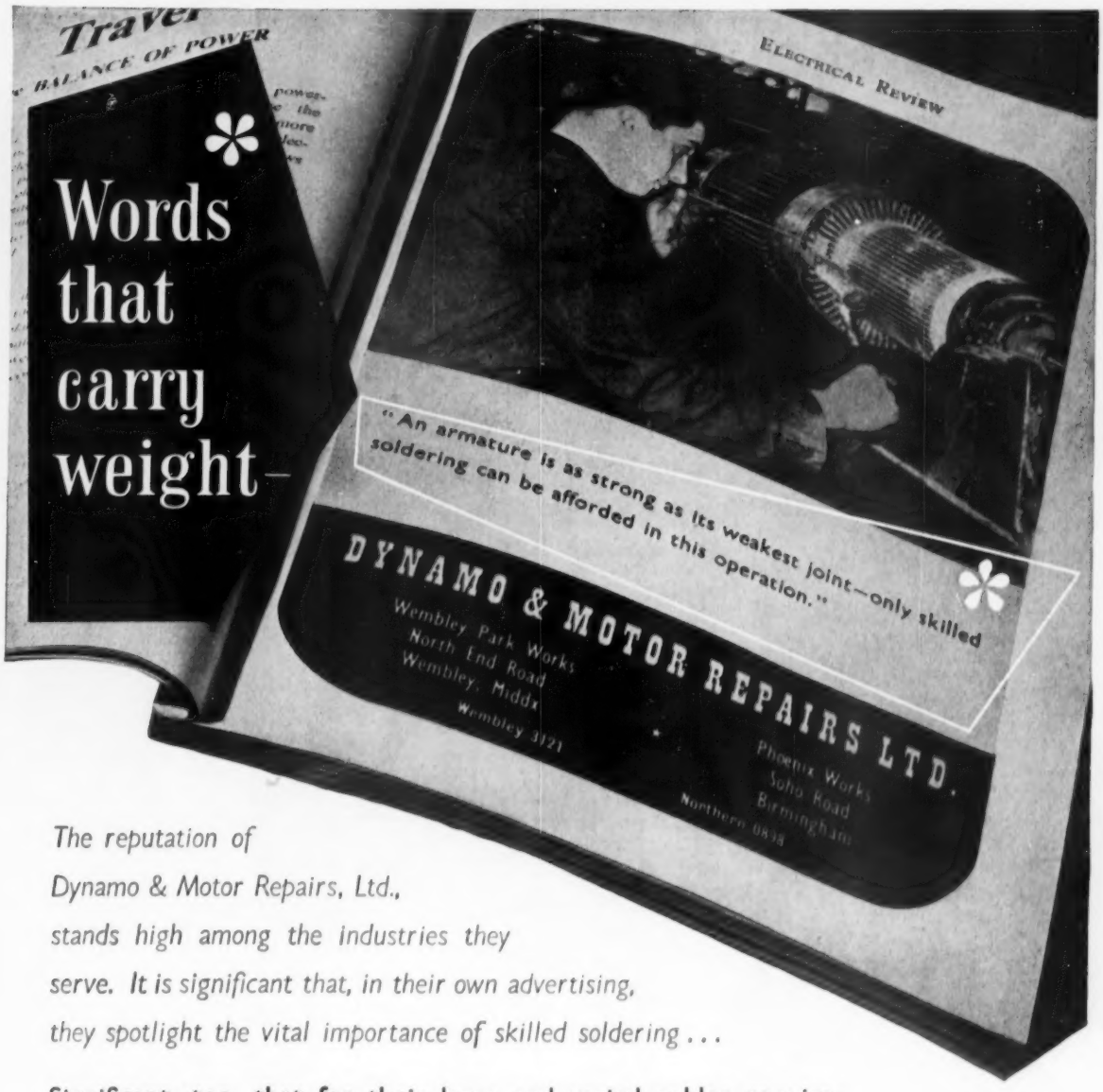
VANDERVELL

BEARINGS AND BUSHES

as original equipment

VANDERVELL PRODUCTS LIMITED · WESTERN AVENUE · ACTON W3 · LONDON

Smee's V 96



The reputation of Dynamo & Motor Repairs, Ltd., stands high among the industries they serve. It is significant that, in their own advertising, they spotlight the vital importance of skilled soldering . . .

Significant, too, that for their large and varied solder requirements they rely on the quality and purity of Enthoven products.

ENTHOVEN SOLDER PRODUCTS

ENTHOVEN Ingot, Plumber's, Tinman's, Blowpipe, Solid Wire and Cored Solders and Fluxes are supplied to B.S. (where applicable) or customers' own specifications. Our technical staff is always available for consultation. Technical data, descriptive literature and samples will gladly be sent on request.

Enthoven Solders Ltd., 89 Upper Thames Street, London, E.C.4 Mansion House 4533

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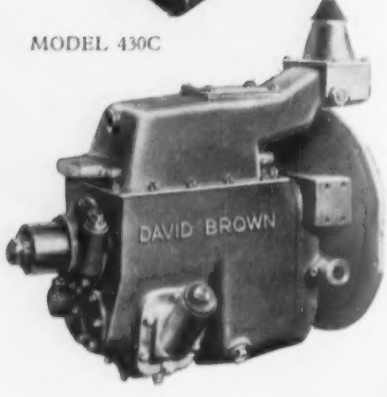
4 AND 5 SPEED GEARBOXES

David Brown gearboxes are backed by over half a century of unrivalled experience in gearbox manufacture, and outstanding features include silent running, ease of operation, simplicity, sturdiness, low weight to strength ratio, compactness and dependability in service.

There is a gearbox in this range to meet your requirements and David Brown engineers will be pleased to offer their recommendations.

Also available as a standard unit is the 2-speed epicyclic auxiliary gearbox, offered as either an overdrive or underdrive and suitable for 350 lb./ft. maximum engine torque.

MODEL 430C

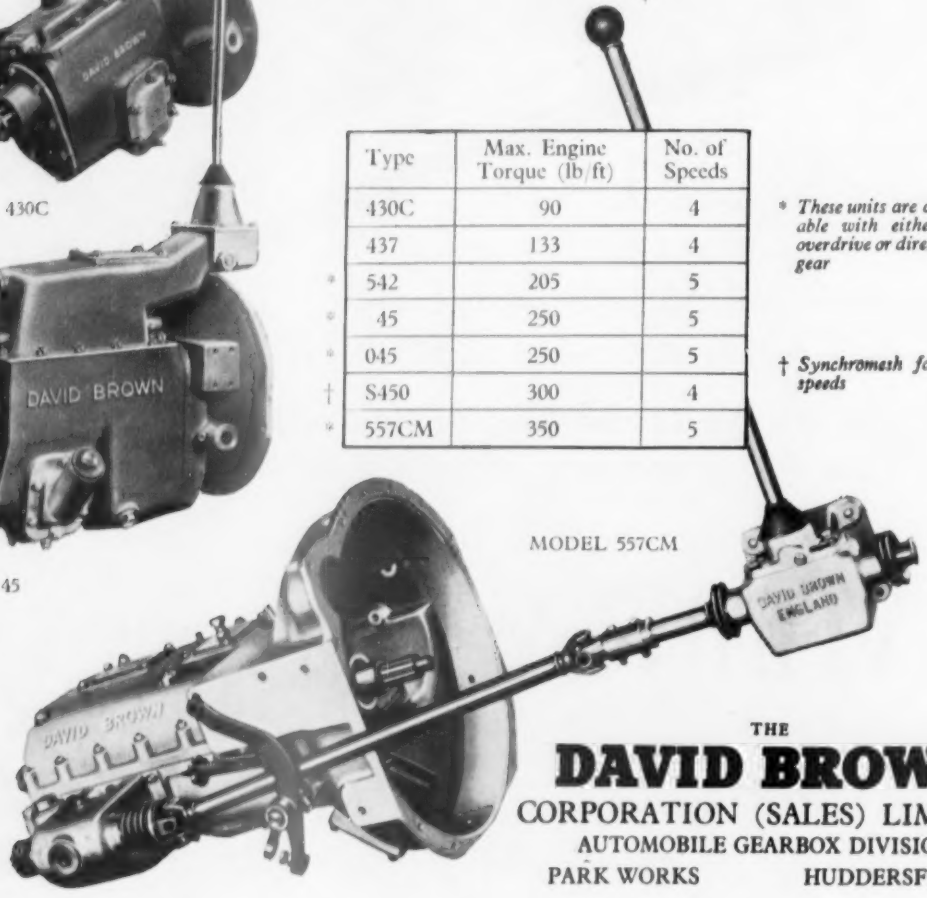


MODEL 45

	Type	Max. Engine Torque (lb/ft)	No. of Speeds
	430C	90	4
	437	133	4
*	542	205	5
*	45	250	5
*	045	250	5
†	S450	300	4
*	557CM	350	5

* *These units are available with either an overdrive or direct top gear*

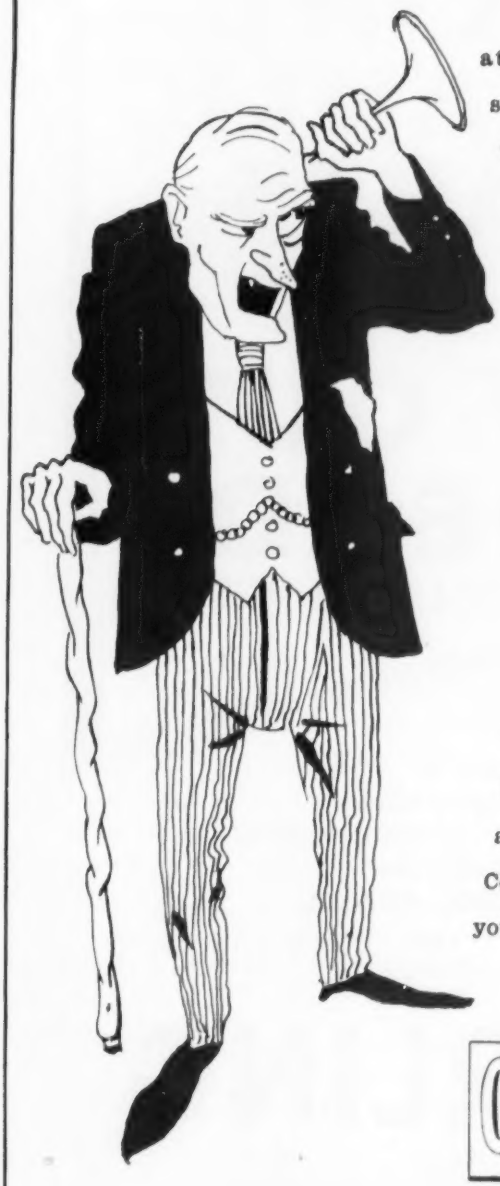
† *Synchromesh for all speeds*



MODEL 557CM

THE
DAVID BROWN
CORPORATION (SALES) LIMITED
AUTOMOBILE GEARBOX DIVISION
PARK WORKS HUDDERSFIELD

Centenarians May Sneer



at a firm with a mere 50 years record of successful selling to the Motor Industry to their credit. For ourselves, we feel modestly proud to have been in at the beginning and even prouder (not to say quite a little relieved) to be in it still. We started our works up in Coventry right in the heart of the new industry when the wheels first began to turn. We have grown up with it — and it gives us a lot of satisfaction to know that some of the first names on our order book are still very good customers—and those names are world-famous wherever cars are made, driven or discussed! What do we make? Trim Assemblies, Carpets, Screens, Hoods, Tilt Covers for lorries & W.D. vehicles, tractor covers and so on. (We have also developed a plastic extrusion and moulding side to the business—if you want anything with knobs on!) Why aren't we better known? Because we've always been a "family" concern and not a large combine—we employ about 150 people. Could we supply you? We should certainly like to know what your requirements are and be given the chance to quote!

Coventry Motor Sundries

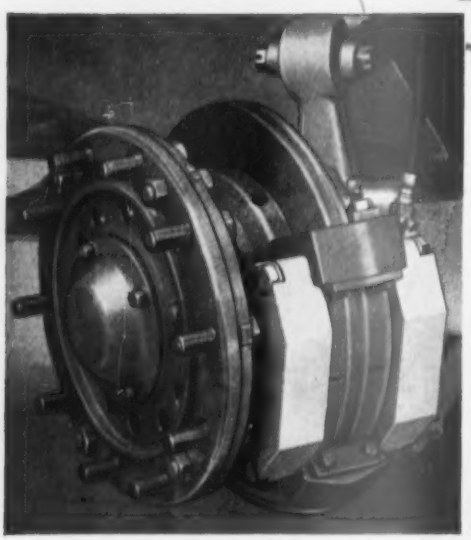
COVENTRY MOTOR AND SUNDRIES CO. LTD.
SPON END, COVENTRY. TEL: COVENTRY 40363/4

Cogent

in terms of modern progress and development

obviously

GIRLING HYDRAULIC BRAKES *are* WAY OUT AHEAD



for the first time in
Passenger Transport History
BIRMINGHAM'S FAMOUS 'MIDLAND RED'
fit as standard equipment
GIRLING DISC BRAKES
to the new S.14 type, 44 seater
Public Transport Vehicle

Striking evidence of progressive development, the supply of Disc Brakes for Public Service Vehicles marks one more milestone in the historic Girling journey into the future.

GIRLING

THE BEST BRAKES IN THE WORLD

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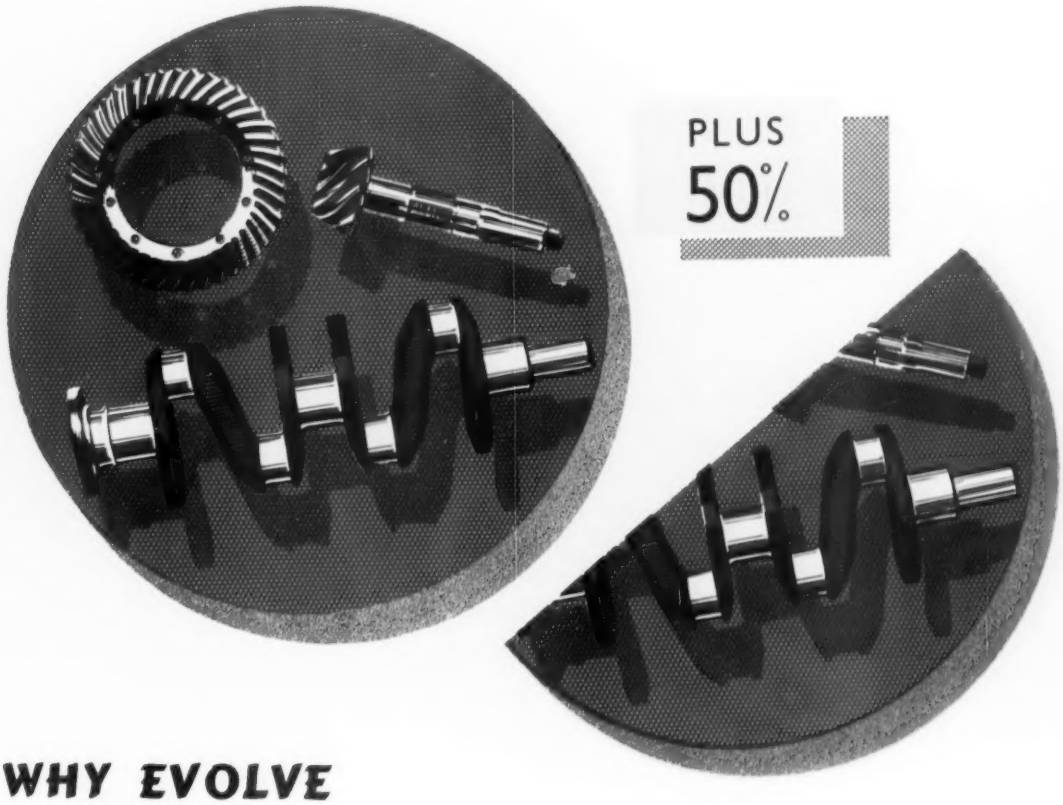
MOLYBDENUM'S PLACE IN INDUSTRY



WHY EVOLVE NEW STEELS?

Among others, the automobile industry plans a vast increase in production, but the supply problem is not solved simply by ordering 50% more of the important alloy steels.
For every one of these steels is dependent upon the alloying elements used in it, and the serious shortage of even one of them prohibits expansion on present lines.

THIS IS WHERE MOLYBDENUM CAN HELP

Molybdenum is an alloying metal chiefly used in small quantities to improve the physical properties of steel.
By increasing the Molybdenum content, however, all the desired physical properties may be attained, and at the same time the scarcer elements reduced.

Molybdenum is abundant and its price is not subject to violent fluctuations. It comes to this country as a concentrate, free from import duty, and we have resources estimated to supply steelmakers for 300 years.

WE HAVE A WHOLE MOUNTAINFUL OF **MOLY**
PRONOUNCED 'MOLLY'

CLIMAX MOLYBDENUM COMPANY OF EUROPE LIMITED, 2-3 CROSBY SQUARE, LONDON, E.C.3
TECHNICAL ENQUIRIES TO 99 PINSTONE STREET, SHEFFIELD, 1

pioneers of progress

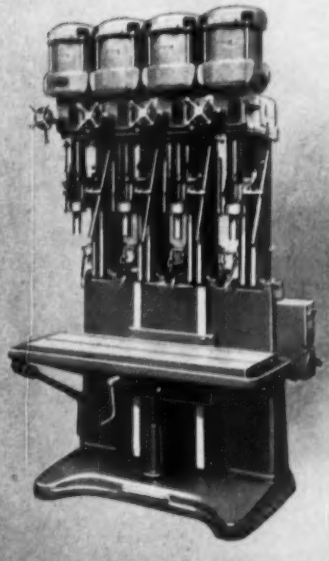
True to the great traditions of Coventry whose long history is closely linked with the progress of Britain . . . Sterling Metals operating large *Aluminium* and *Cast Iron* foundries and, as the world's largest producers of "*Elektron*" *Magnesium Alloy* castings, have built up an unequalled reputation for skill, quality and craftsmanship.

STERLING METALS LTD

COVENTRY

HERBERT

TYPE M MULTI-SPINDLE. MAX. OF 12 SPINDLES. FOUR SPEEDS 540 to 2,900 r.p.m.



TYPE V HAND-FEED GEARED SPINDLES. DRILLS and TAPS UP TO 1¼". ALTERNATIVE RANGES OF SPEEDS. MAXIMUM 2,850 r.p.m.

ALL-ELECTRIC DRILLING MACHINES

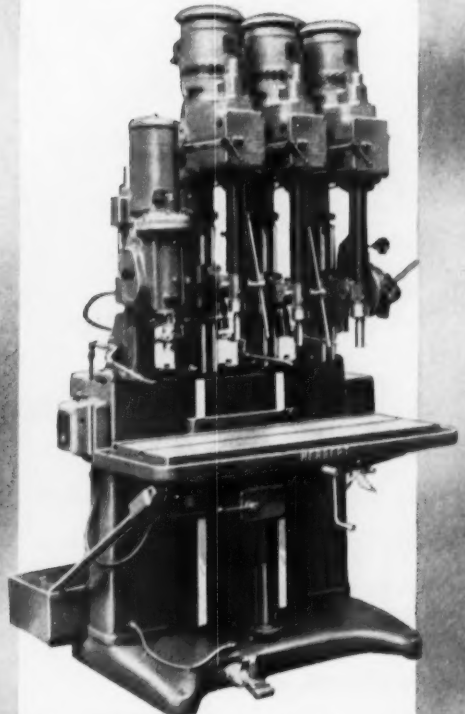
For all drilling and tapping operations up to 1¼".

Easy to install.

Easy to operate.

Instantaneous and effortless control.

Unit construction and interchangeability enables a wide range of special requirements to be met.

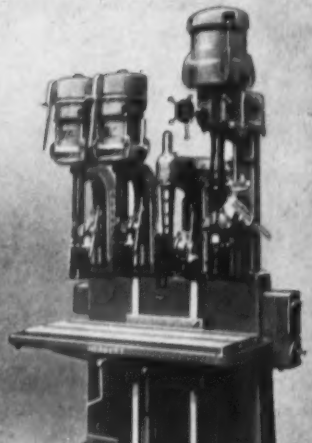


THREE TYPE C TOP COLUMNS AND FLASH TAPPER ON A SINGLE BASE

EARLY DELIVERIES

INCREASED RADIAL CAPACITY ON TYPE V

TYPE B BENCH. SPEEDS UP TO 18,000 r.p.m.



COMBINATION OF TOP COLUMNS MOUNTED ON A SINGLE BASE. SPEED RANGE 74 to 5,600 r.p.m.

ALFRED HERBERT LTD · COVENTRY

How much does it cost to drive a screw? The Phillips cruciform recess accurately fits its special driver for fastest, safest, easiest and tightest assembly. With Phillips screws you can reduce your assembly costs often by as much as 50%, but with Linread Phillips screws you will obtain all these advantages and the additional product appeal and savings which come from using the best, both in quality and service.

Linread

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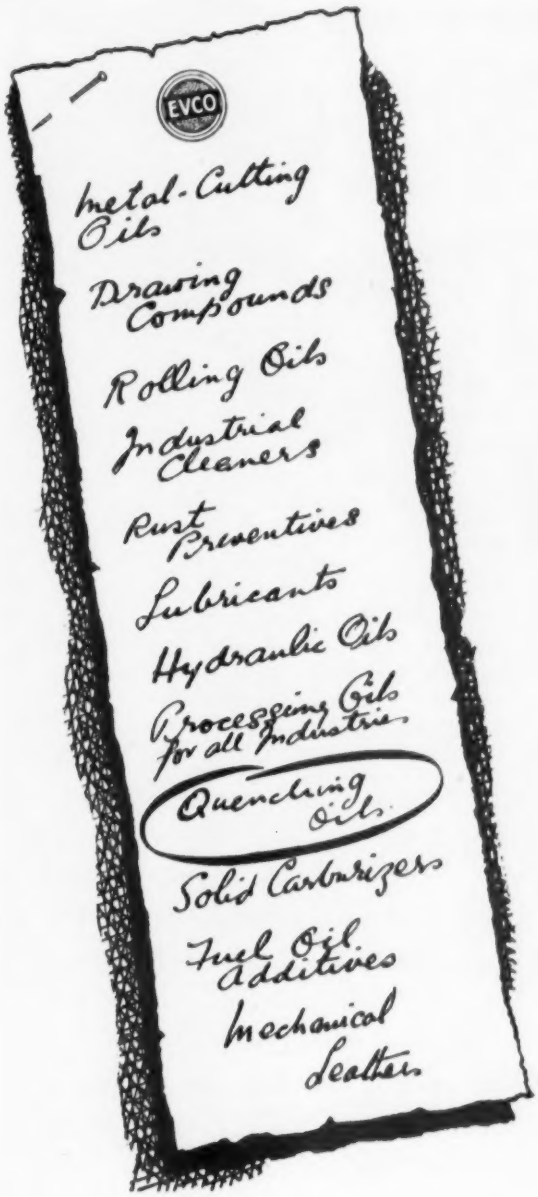

QUENCHING OILS

No. 2 SOLUBLE QUENCHING OIL for uniformity and years of service.

No. 2 Soluble Quenching Oil stands pre-eminent in providing uniform quenching speed and years of service throughout a wide temperature range.

It *remains* uniform with continued use and is good to the last drop.

No. 2 Soluble Quenching Oil outlasts other quenching oils and based on ultimate cost, is the most economical to use.

We have a demonstration Heat-Treatment plant at our Birmingham works and our metallurgical staff will gladly advise on the selection and best application of our efficiency products.

Edgar Vaughan & Co. Ltd.

BIRMINGHAM 4 · ENGLAND

Works and Depots at: Birmingham, Manchester Liverpool, Southall (Middx.), Bristol, Glasgow

For high efficiency at low speeds

E.N.V. Double Reduction drives and axles are extensively used for battery-electric vehicles of all types, for petrol and diesel engined industrial trucks, dumpers and special purpose low-speed vehicles. Primary reduction is by helical gears and final drive by spiral bevels ensuring high efficiency over the entire speed range.

A range of standard gears provides alternative overall ratios from 8 : 1 to 19 : 1. Standard types of axles are suitable for vehicles up to 2 tons total rear axle load. Nosepiece assemblies incorporating double reduction drive and differential are available in three sizes for building into mechanical handling equipment.

E.N.V. WILL BE GLAD TO CO-OPERATE IN THE DESIGN OF TRANSMISSIONS FOR YOUR NEW PRODUCTS.

ENV for gears

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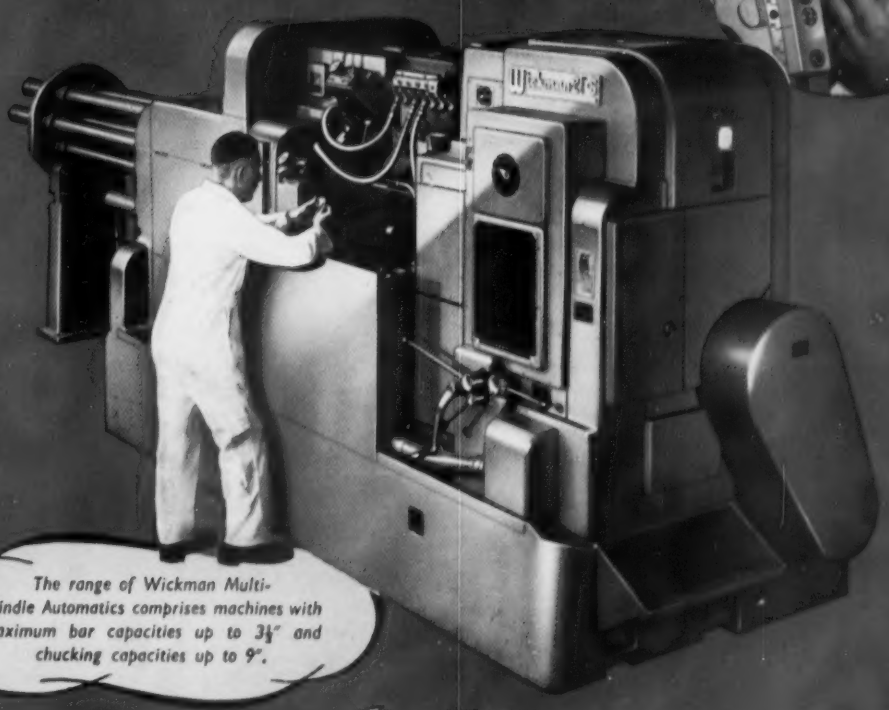

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Turin Show

FOR those whose business it is to study the coachbuilder's art, the Turin Motor Show is probably the most interesting of the Continental exhibitions. Superficially, it may appear that vehicles manufactured in small quantities will show little of interest to designers concerned with large quantity production. This is not the case, however. When a model has to be continued for a number of years, as is the case in quantity production, evolution in design is necessarily much slower than when only a few vehicles are produced from one drawing and new designs follow in relatively quick succession. It is, therefore, understandable that progress in certain directions can more readily be made by coachbuilders.

Although any marked changes in body styling are usually initiated by coachbuilders, and are taken up by the larger concerns only after they have been proved acceptable to the public taste, this year there were no outstanding departures from accepted practice. There were, however, many interesting detail developments.

Ventilation

Ventilation is still far from satisfactory on many cars; because of this the method adopted by Ghia, of Turin, for ventilating a body mounted on a Ferrari chassis, is of interest. A vertical louvre is formed above the waist, immediately behind the door shut-pillar. It faces rearwards, so that the tendency is for air from inside the body to be extracted through it. A simple shut-off control is fitted inside each rear quarter. The need for a ventilator of this type, fitted either in conjunction with or as an alternative to a hinged ventilation panel on each door, is being increasingly recognized; in fact, there were other examples to be seen at the show.

Hinged ventilating panels at the front of the door lights have certain disadvantages; they are not easy to make watertight, and their glazing frames tend to obstruct the driver's field of vision. Therefore, the incorporation of ventilators in the rear quarters might be an attractive alternative, provided they can be controlled from the driver's seat. Moreover, as most of the blowers used in heaters do not operate effectively against a back pressure, the best results can be obtained from them only if there is an outlet through which the stale air can be exhausted so that fresh heated air can come in to take its place.

There was strong evidence that the two-door, four-light type of body wherein the door shut-pillar extends up from the sill only as far as the waist rail, is becoming more popular. There were many American exhibits of this type, and Pinin Farina also displayed a body of this form on a Lancia chassis. The advantages of this arrangement include improved arc of vision and reduced weight. This is also a very good layout for tropical climates, since it protects the occupants of the car from the sun and at the same time provides ample ventilation from each side. In other words, it is a good compromise between the saloon and convertible types of design.

Sidelights

Another interesting feature of the Pinin Farina body on the Lancia chassis is the curved glass side lights. A winder mechanism is used to raise and lower each light into its housing below the waist. The rear quarter lights move backwards and downwards, instead of straight down, to break the seals at their leading edges and to clear the shut-pillar and lock mechanism. In the raised position, all four lights seal directly against a rubber strip on the screen and rear pillars and cant rail. The wrapped-round windscreen arrangement adopted on this vehicle allows glass of approximately rectangular form to be used for each door light.

On low-built sports cars, the positioning of the exterior handle on the door presents some problems that are not easily solved; this is shown by the arrangement recently adopted for one vehicle in fairly large scale production. Therefore the arrangement on a Vignale Demon Rouge on a Fiat V8 chassis is of interest. A trigger type release, large enough to be actuated by the whole hand rather than by one finger, is fitted on each side at the forward edge of the wrapped-round rear light. Although it actuates a link immediately behind the shut-pillar, the lock mechanism is housed in the usual manner. The link bears against a push rod, which is attached to the latch, and which projects through the rear edge of the door inner panel. When the link is actuated by the trigger, it pushes the rod, and therefore the latch, forward to unlock the door.

In many quarters, even in the United States of America, there is, nowadays, a tendency to regard Italian coachbuilders' styling as *hors concours*; most definitely we cannot subscribe to this opinion. We continue to consider the more restrained British coachbuilt styles superior. Moreover, it is very doubtful whether those engaged in quantity production find any marked inspiration in the Italian developments that attract the most attention. Indeed,

more often than not, the attention is caused by the novelty of the style rather than by any positive virtues that it embodies. Detail developments, however, of the character briefly discussed here, are a different matter, and a study of Continental practice will often be well repaid.

Road Safety

ROAD safety is one of the many problems that are studied at the Road Research Laboratories. The investigations are mainly concerned with highway engineering, but they also cover some aspects of automobile engineering. For instance, the work on skidding, in addition to the major concern with road surface materials, has also included a study of the behaviour of a vehicle fitted with a device to prevent the locking of the front wheels when the brakes are fully applied. The results obtained and recorded in the form of a film are most impressive.

Two tests were carried out with a passenger car. In each test the driver applied the brakes hard, as in an emergency, and at the same time attempted to steer a zig-zag course to avoid hitting obstacles placed in the path of the car. When the vehicle was fitted with the device for preventing the front wheels from locking, steering control was maintained and the vehicle stopped in a remarkably short distance among the obstacles, but without hitting any of them. When the device was not fitted, the wheels locked and the vehicle continued in a straight line, knocking down the obstacles, until it finally came to rest.

There is no doubt that very many accidents would be avoided if the driver had complete steering control while braking hard in an emergency. The device used at the Road Research Laboratories has been designed for aircraft and therefore is not entirely suitable for application to road vehicles. There seems to be no reason, however, why a mechanism could not be developed for this purpose. Essentially, the requirement is that the brakes shall be released slightly when relative movement between the drum and the back-plate ceases. In hydraulic brake systems, this presumably could be best effected by releasing some of the hydraulic pressure the instant the wheels lock.

In addition to the fact that this type of device allows steering control to be maintained, there is the added advantage that maximum braking is also obtained. This is because the best possible results are experienced only if the wheels are almost at the point of locking when the brakes are applied hard; if they actually lock, the coefficient of friction between the road and the tyre is reduced.

Other investigations carried out at the Laboratories include the testing of motor cyclists' crash helmets. At first sight this may not appear to be of interest to automobile engineers. The results, however, may have an indirect bearing on the design of crash pads for dash facia panels and screen head-rails in sports cars. Rather surprisingly, the tests have shown that relatively hard cork is much better than soft rubber as a lining for crash helmets; which shows the folly of jumping to hasty conclusions about the suitability of different classes of material for this type of application.

As the conditions are different, further tests would no doubt be necessary to determine whether the results obtained for crash helmets are also applicable to crash pads for motor vehicles. Whereas a crash helmet is designed to be effective as a buffer between the head and a hard unyielding road surface, the crash pad in a car, in most instances, is mounted on a panel that has itself a certain degree of resilience.

Investigations have proved that a large proportion of serious injuries arising from motor accidents occur when the passengers are thrown out of the vehicle, usually when the doors have been forced open by the impact. In general, injuries are less severe when the occupants are retained in the vehicle. This may indicate that accessory manufacturers could find a market, particularly among sporting motorists, for safety bolts for doors, and for straps to keep the occupants in their seats. If straps are fitted, the attachment of the front seats to the floor requires attention. In almost all vehicles of current design, the existing attachment arrangements are inadequate to withstand impact forces of the severity that will probably be experienced in crashes.

A comprehensive study of the tendency for the driver to suffer injury from the steering wheel should also be made. For example, it is not clear whether, in a crash, the wheel is forced back and hits the driver, or whether the driver is forced forward on to the wheel. It would also be helpful to know if it is the rim or the hub of the wheel that causes the injury. Until such data are available, satisfactory counter measures to prevent, or to reduce, the severity of such injuries are scarcely feasible.

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A.E.C. RELIANCE CHASSIS

A Public Service Vehicle Chassis Designed for Economy of Operation

IN the development of the A.E.C. Reliance chassis the aim has been at producing at a relatively low cost a vehicle that can be operated for long periods with the minimum of servicing attention. It has been designed to take bus or coach bodywork with provision for 41-44 seats. The laden weight of the complete vehicle is under 9½ tons. This light weight has been obtained by efficient design without any sacrifice of strength or stiffness.

All the major mechanical components were conceived initially to suit the whole range of vehicles, that is, the Reliance bus and coach chassis, the Monocoach chassisless vehicle and the Mercury goods carrying vehicle. For the Mercury chassis, the vertical, AV410 and AV470 units have been developed, because the underfloor layout is unsuitable for tankers and tractors, and presents difficulties when tipper bodies are required to be installed on the chassis. In the Monocoach and Reliance, either the AH410 or AH470 horizontal engine is installed. These differ only slightly from the vertical units: the main differences are in the shape of the sump and crankcase extension, the arrangements of the dynamo drive and mounting, the engine mounting brackets, the exhaust manifold, oil filler tube and fan drive.

Adoption of the horizontal layout for the Reliance has enabled a flat saloon floor to be obtained over the whole length of the chassis, and the weight distribution over both axles is favourable. The fact that from the outset all the components have been designed specifically for the whole range has meant that the development of each of the vehicles has been carried through to a logical conclusion without recourse to undesirable makeshift arrangements such as are sometimes adopted when existing mechanical components have to be used in a new vehicle design.

Because of the light weight of the vehicle, it has been possible to employ a 7·68 litre engine, the AH470 unit, instead of the 9·6 litre engine of the earlier A.E.C. models. This, the manufacturers claim, gives a fuel consumption of 14-18 m.p.g. instead of the 9-12 m.p.g. of the earlier design. In fact, it would appear that A.E.C. are modest in making these claims.

In a road test report published in the 13th August 1954 issue of our associated journal, *Motor Transport*, it is estimated that a consumption of

SPECIFICATION

TRANSMISSION: Single dry plate clutch, with sprung centre. Gear ratios: top 1:1, fourth 1·56:1, third 2·65:1, second 4·4:1, first 6·25:1, and reverse 6·01:1. Propeller shaft: Hardy Spicer type 1600, with needle roller bearing universal joints. Rear axle: Single reduction, fully floating spiral-bevel, with alternative ratios of: 4·7:1, 5·22:1, 5·87:1 and 6·28:1.

FRONT SUSPENSION: Reversed Elliott type axle with leaf springs and Armstrong AT11 double-acting, hydraulic shock absorbers.

REAR SUSPENSION: Through-axle with semi-elliptic leaf springs.

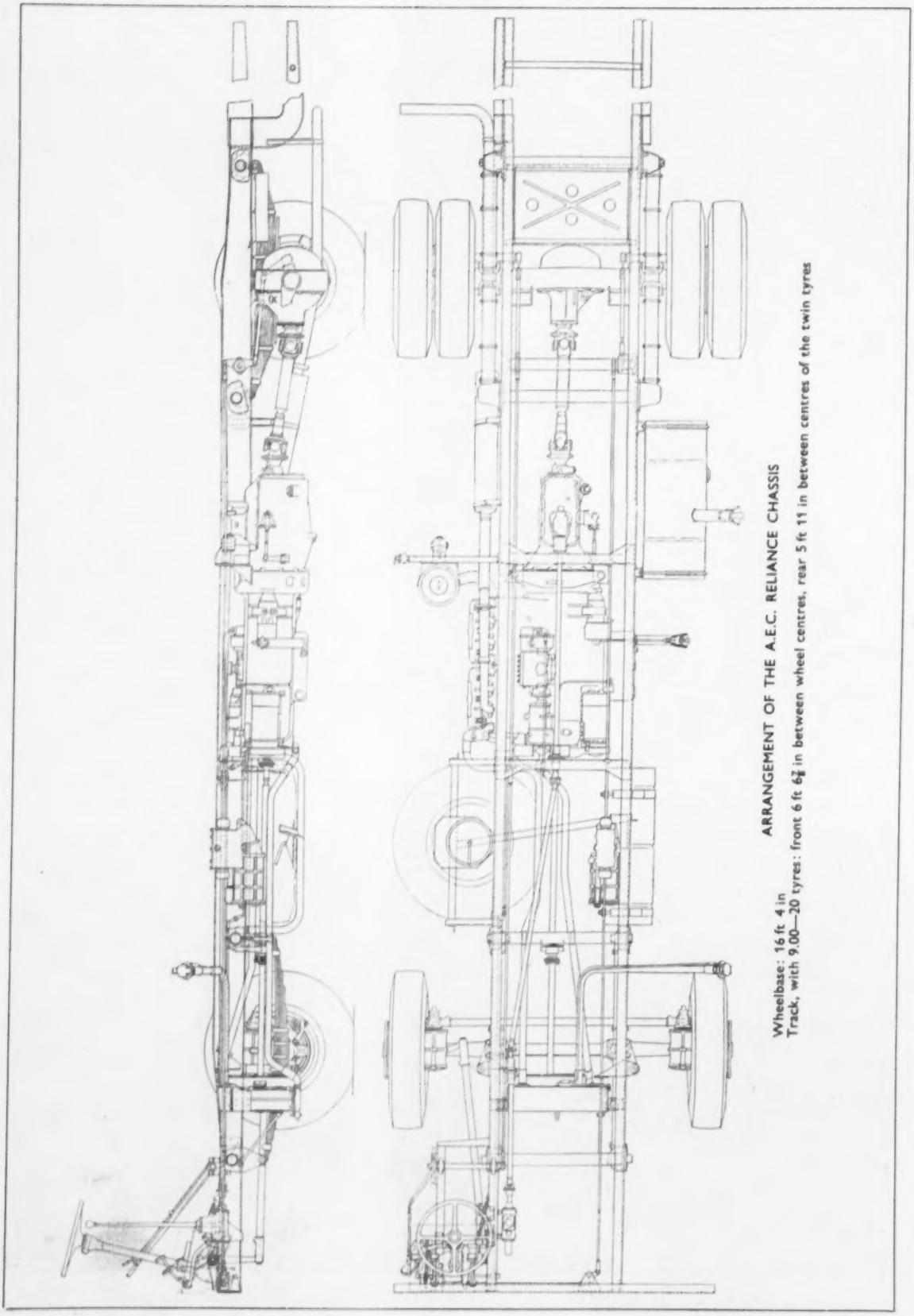
STEERING: A.E.C. worm and nut type with a ratio of 32:1, giving 5½ turns from lock to lock. Steering wheel, 20 in diameter. Turning circle: 62 ft.

BRAKES: Girling, cam-operated leading and trailing shoe type. Drum diameter: 15½ in. Shoe width: front 4½ in, rear 6 in. Shoe area: Front 256 in², rear 362 in².

TYRES: Single front and twin rear, 9·00—20, 10 ply, and for export 10·00—20, 12 ply. Pressure: 65 lb/in². Three-piece welded type wheel.

DIMENSIONS: Wheelbase 16 ft 4 in. Track: With 9.00—20 tyres, front 6 ft 6½ in between wheel centres, rear 5 ft 11 in between centres of twin tyres. Ground clearance, laden: At rear axle casing 9 in, at radiator, gearbox and engine 10 in. Overall length: 29 ft 6½ in. Overall width: 8 ft 0 in. Mean height of top of frame from ground: 2 ft 9½ in, with 9.00—20 tyres. Front overhang: 5 ft 1½ in. Rear overhang: 8 ft 1½ in. Dry weight, without spare wheel and carrier: 3 ton 8 cwt. Engine dry weight, with exhauster, front mounting bracket, dynamo and starter, but without oil, clutch and flywheel: 1,185 lb. Gearbox weight: 273 lb. Clutch weight: 98 lb.

On the A.E.C. Reliance, diagonal ties are employed to brace the frame in front of and behind the engine bay



ARRANGEMENT OF THE A.E.C. RELIANCE CHASSIS

Wheelbase: 16 ft 4 in
Track, with 9.00—20 tyres: front 6 ft 6⅞ in between wheel centres, rear 5 ft 11 in between centres of the twin tyres

17-18 m.p.g. should be attainable on an average long distance service with a fully laden coach. Actually, a figure of 19·4 m.p.g. was obtained on one of the test runs carried out at an average speed slightly higher than 31 m.p.h.

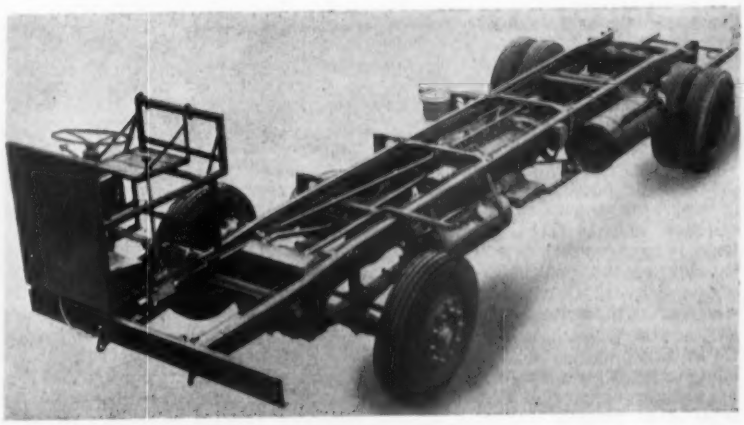
Where operating conditions are suitable and operators require even greater economy, a smaller engine can be fitted. This is the AH410, 6·75 litre unit. In some instances, it might be advantageous to operate a mixed fleet comprising some vehicles with the smaller and others with the larger power unit, according to the duty they are required to perform. This is practicable economically because most of the engine components are common to the whole range. The engines were described in detail in the October 1954 issue of *Automobile Engineer*.

Briefly, the specifications of the power units are as follows. Both are six-cylinder, direct injection oil engines with a maximum governed speed of 2,000 r.p.m. For the bus chassis, to obtain greater fuel economy, the maximum engine speed is restricted to 1,800 r.p.m. However, the higher speed setting can be retained if required by the operator.

The AH410 unit, as its designation implies, has a swept volume of 410 in^3. Its bore and stroke are 4·13 and 5·12 in respectively. At 2,000 r.p.m., the power developed is 98 b.h.p., while 90 b.h.p. is developed when the engine is governed at 1,800 r.p.m. The maximum torque is 282 lb-ft at 1,300 r.p.m.

On the other hand, the AH470 unit, with a capacity of 470 in^3, has a bore and stroke of 4·41 in and 5·12 in respectively, and develops 112 b.h.p. at 2,000 r.p.m. When the engine is governed at 1,800 r.p.m., the maximum power output is 103 b.h.p. The maximum torque is 325 lb-ft at 1,100 r.p.m.

Although the five-speed gearbox is the standard unit and is common to the whole range of vehicles, the Reliance can be supplied with the A.E.C. Monocontrol, electro-pneumatic, two-pedal arrangement if required. The single-speed, spiral bevel back axle is also common to the whole range, but four different ratios are available.



The Clayton Dewandre vacuum servo brake operation system is the standard installation for the Reliance chassis

So far as possible, the control movements have been arranged so that they are similar both as regards distance moved and the force required to actuate them. The manufacturers maintain that it is fundamentally wrong to expect the driver to apply, for example, a light force to the clutch pedal and simultaneously a heavy thrust to the brake pedal. Likewise, the gear shift movement and force required to effect it should be comparable with that of the other controls. Rationalization of control forces and movements contributes much to reduction of driver fatigue. The steering of this vehicle is exceptionally light. In fact, during the development stage, some extra damping had to be incorporated. The brakes are operated either by the Clayton Dewandre triple, vacuum servo system or by compressed air.

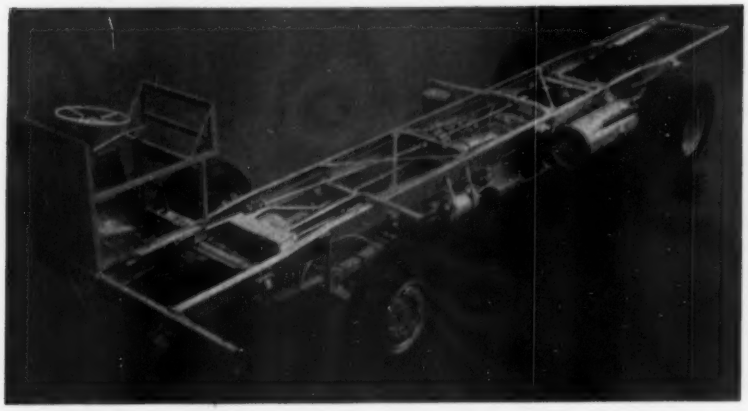
Clutch and gearbox

A 15½ in diameter single dry plate clutch, with a sprung centre, is employed. The friction lining area is 237 in^2. Fifteen presser springs are fitted and the spring load assembled is 1,875 lb. There is no compensating device or engine tie because hydraulic actuation has been adopted. This type of actuation obviates the need for adjustment to compensate for wear of the friction linings. A ball thrust bearing is employed in the actuation mechanism. The whole unit is enclosed in a bell-housing integral with the magnesium alloy gearbox.

The five-speed gearbox weighs 273 lb dry. It holds 2 gallons of oil; S.A.E. 140 is recommended. The gear ratios are: top 1:1, fourth 1·56:1, third 2·65:1, second 4·4:1, first 6·25:1, and reverse 6·01:1. An inertia-lock type synchromesh unit is employed on all gears except first and reverse, which are of the sliding mesh type.

At the spigot bearing, the En 39A primary shaft is about 1 in diameter. Immediately behind this bearing, the shaft is threaded for a nut that pulls the hub of the centre plate of the clutch on to a taper of 1 in 8 on the diameter. A plain key transmits the drive. Behind this, the shaft is 1¾ in. nominal diameter where it passes through a sleeve extension of the B.S.1452 grade 12 casting that houses the lip type front oil seal and retains the single-row roller bearing immediately in front of the primary gear. The two-piece inner race of the bearing is pulled against the gear by a nut on a 1¼ in diameter thread round the shaft. A flanged, cupped, mild steel housing carries the outer race of the bearing, which is held in position by the spigoted-in sleeve casting, already mentioned. This arrangement has been adopted so that the primary gear can be withdrawn through the hole in the front wall of the box. The whole assembly is retained in the bore in the end wall of the box by five ⅜ in diameter studs. The inside diameter of the cupped housing is 100 mm and its outside diameter is 4⅞ in.

Formed at the rear end of the primary shaft is the primary gear and an extension that carries the En 34 synchro cone, which is pressed on and keyed. The roller type spigot bearing for the 25 mm diameter front end of



The A.E.C. Reliance with the air pressure braking installation, which can be supplied as an alternative to the vacuum servo system

the mainshaft is in a 62 mm counterbore in the primary shaft. The teeth of the female component of the toothed clutch of the synchro unit are machined in this counterbore. Drilled from the front face of the primary gear into the counterbore are a number of $\frac{1}{4}$ in diameter holes, through which a drift can be passed to knock out the bearing for servicing. A flanged distance ring is fitted between the inner race of the bearing and a shoulder on the mainshaft.

Simplicity, robustness and compactness are the outstanding features of the synchronizers. At the front, the En 34 centre part of the unit, the male component of the toothed clutch, is appreciably smaller in diameter than most other designs. Its teeth for the transmission of direct drive have a pitch circle diameter of 2·500 in, while those for fourth speed gear are of 3·571 in pitch circle diameter. The component is mounted on a 2 in diameter splined portion of the mainshaft. There are ten splines, approximately $\frac{3}{16}$ in deep.

Six holes, drilled radially in the sliding, centre component, house $\frac{1}{4}$ in diameter En 36T pegs. These pegs project through slots in the En 39A sleeve component. Their outer ends register in holes in the En 34 selector collar, which is a separate component. Three more $\frac{1}{4}$ in diameter, radially drilled holes house the spring-loaded ball detents. An unusual feature is that a $\frac{1}{16}$ in diameter hole is drilled from the rear end face of the component to communicate with a $\frac{1}{32}$ in diameter axial hole drilled from the base of the detent housing: this is to obviate any possibility of the detent balls being locked hydraulically by oil in their housings.

The phosphor bronze synchro cups are pressed into the ends of the sleeve and secured by six $\frac{1}{16}$ in diameter countersunk rivets. They have forty-four $\frac{1}{32}$ in wide slots machined parallel to the axis of the unit in their conical faces, which are grooved peripherally. This allows the cones to break rapidly through the oil film during engagement and prevents their sticking during disengagement: the slots act as reservoirs for the oil squeezed from between the conical faces as they engage. The mean diameter of the 13 deg conical faces of both the top and fourth speed units is about $4\frac{7}{16}$ in, and the length of engagement between cone and cup is $\frac{1}{2}$ in.

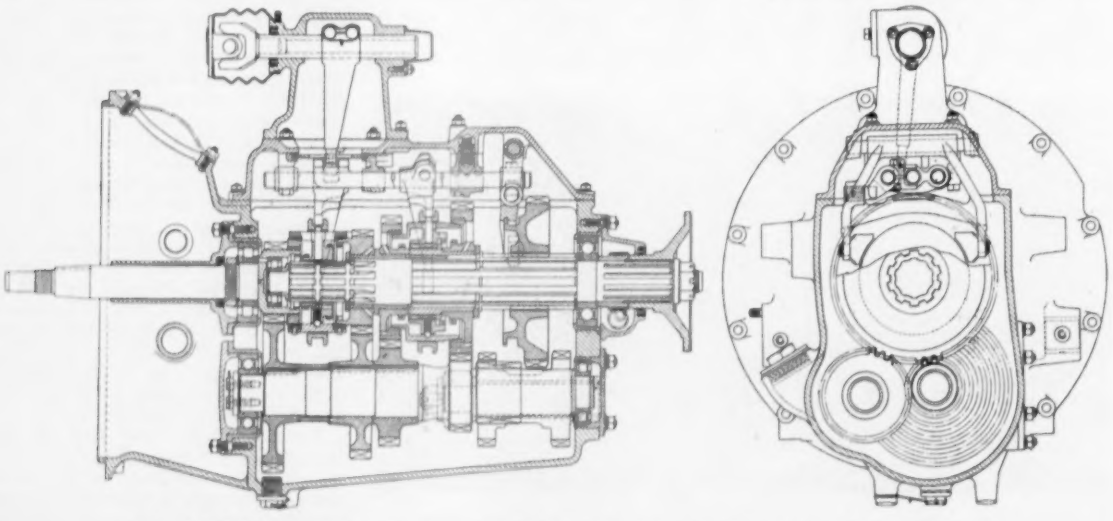
In the sleeve, the slots through which the pegs in the centre component project are machined parallel to the axis of the unit. A U-shaped notch, large enough to accommodate the peg, is cut on each side of the centre of each slot. The operation of the synchro unit is as follows. When a gear is selected, the whole assembly slides axially on the mainshaft until the cones engage. If the driving and driven assemblies are not synchronized, the centre component rotates relative to the sleeve until the radial pegs engage in the U-shaped notches, thus baulking further axial movement. When synchronization has been effected, the pegs are free to ride out of the notches, so the spring-loaded balls in their detent grooves can be over-ridden and the centre component of the toothed clutch moved into engagement.

Immediately behind the synchro unit, the mainshaft is shouldered to $2\frac{1}{32}$ in diameter to carry the phosphor bronze bush for the fourth and third speed gears. The bush is located against rotation relative to the shaft by a key, the end of which registers in a slot in the rear end of the bush. This key is carried in a slot in the larger diameter portion of the shaft behind the bush. A $\frac{1}{32}$ in thick flange round the centre of the bush separates the fourth and third speed gears and the whole assembly is retained by a $\frac{1}{4}$ in thick, internally splined washer in a groove immediately in front of the shoulder where the shaft is reduced in diameter to carry the synchronizer unit.

This washer is assembled over the splined end of the shaft and then rotated in the groove until its splines are out of normal phase relative to those on the shaft. Then two keys are inserted between diametrically opposite pairs of splines on the shaft and pushed to the rear to register between the splines in the washer. These keys have T-heads that seat in a groove round the shaft, which locates them axially. The bearing surface for the fourth speed gear is about $1\frac{7}{16}$ in long × $2\frac{7}{16}$ in diameter, and that for the third speed gear is about $1\frac{1}{4}$ in long × $\frac{11}{16}$ in diameter. Holes, $\frac{1}{32}$ in diameter, are drilled from the adjacent faces of each gear to allow oil to be splashed in to the bearing surfaces.

The synchro cone for fourth speed is integral with the gear. On the other hand, the cone for the third speed gear is formed on the sleeve component of the synchronizer, and the phosphor bronze cup is housed in the gear. This cup is held in position by three $\frac{1}{16}$ in diameter countersunk rivets, in holes drilled parallel to the axis of the unit, and with their axes at the interface between the outer periphery of the cup and its housing; in other words, each hole is half in the cup and half in the housing. The male component of the toothed clutch is formed round the boss of the third speed gear, and the centre component of the synchro unit forms the female component. A diametral pitch of 3·571 in has been adopted for the third speed toothed clutch, while that for the second speed clutch is 4·00 in.

Apart from the fact that the cones for third and second speeds are formed on the sleeve, and the centre of the synchronizer forms a female, instead of a male, component of the toothed clutch, this synchronizer unit is similar to that for top and fourth speeds. The centre component slides on a splined En 39A sleeve on the $2\frac{7}{16}$ in diameter portion of the mainshaft. Inside the sleeve there are ten splines, 0·118 in



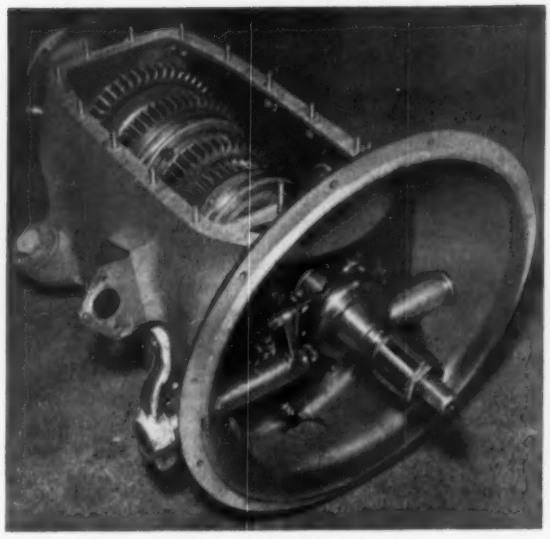
In the A.E.C. Reliance gearbox, the compactness of the synchronizer assemblies is a noteworthy feature

deep, and externally there are twenty-five splines, 0·227 in deep. The sleeve also acts as a distance piece between the rear end of the third and fourth speed gear assembly and a ¼ in thick washer that is splined on the shaft and bears against a collar round the shaft. This collar is immediately behind the second speed gear. The rear portion of the sleeve is shouldered to 2·963 in to form the bush for the second speed gear. Lubrication is provided for by holes drilled through the boss of the gear to the bearing surface.

The male component of the toothed clutch is formed on a forward extension of the boss of the gear. This gear also carries the phosphor bronze synchro cup, which is secured in a similar manner to that in the third speed gear. The mean diameter of the conical face for the third speed synchronizer is 4⅛ in, while that for second speed is 5⅜ in. The length of engagement is ½ in for both, and the surfaces of the cups are slotted and grooved in the same manner as those for top and fourth speeds.

Behind the collar that is formed round the splined portion of the shaft to locate the second, third and fourth gear assembly, is the sliding gear. The boss of this gear is 1⅛ in long and the collar for the selector fork is machined round it. In the rearmost position, it engages the reverse idler, and when it is slid forward, it meshes with the first speed layshaft gear.

To the rear of this, the shaft is reduced to 2 in diameter where it is carried in the ball bearing that axially locates the assembly. A $\frac{1}{32}$ in thick distance washer is fitted between the inner race of the bearing and the shoulder on the shaft. Behind this race are the speedometer gear and the companion flange for the universal joint. The whole assembly is retained by a slotted nut on the 1¼ in diameter threaded end of the shaft. The outer race of the bearing is carried in a $\frac{1}{32}$ in thick flanged, cupped housing inserted into a hole in the crankcase wall. The flange seats in a shallow counterbore round the hole and the whole assembly is retained by an aluminium alloy cover bolted to the rear face of the box. A lip type oil seal housed in the rear end of the cover bears on the boss of the companion flange. The distance between the centres of the bearings in the front and rear walls of the box is about 17 in.



A ball thrust bearing is employed in the hydraulically actuated, clutch withdrawal mechanism

The first and second speed layshaft gears are integral with the En 39A shaft. All the others are assembled on from the front and rear, and the drive is transmitted by plain keys. At the constant mesh gear the diameter of the shaft is 2⅛ in, at the fourth speed gear it is 2$\frac{11}{16}$ in and at the third speed gear 2⅜ in. It is 2½ in diameter at the centre and 2$\frac{29}{32}$ in in front of the rear bearing.

At the extreme front end, the shaft is reduced to 45 mm diameter where it is carried in the ball bearing that axially locates the assembly. A $\frac{1}{32}$ in thick distance washer is interposed between the inner race and a shoulder on the shaft, and the whole assembly is retained by a ¼ in thick plate secured by two ⅜ in diameter set bolts in holes tapped in the end of the shaft. The 100 mm outside diameter outer race is housed in a $\frac{1}{32}$ in thick flanged, cupped housing retained by an end cover spigoted in to the bearing housing and bolted to the front face of the box.

A roller bearing carries the rear end of the shaft, which is shouldered to 40 mm diameter. The inner race of the bearing is retained between the shoulder and a Seeger type circlip in a groove round the rear end of the shaft. The 80 mm outside diameter outer race is simply pressed into its housing, which is flanged, and is spigoted into the rear wall of the box, to which it is bolted. A limited amount of axial float of the inner race and roller assembly is allowed to accommodate axial movement due to thermal expansion.

The nominal diameter of the En 32B reverse idler spindle is 1½ in. It is supported at one end in a hole in the rear wall of the box, and at the other in a boss cast integrally in the side of the unit. A shoulder is formed near the front end of the shaft, which is threaded ¼ in diameter, and a slotted nut is assembled on together with a ¼ in thick washer to pull the shoulder up against the boss.

The idler is carried on two, 1¼ in outside diameter, phosphor bronze bushes inserted one from each end. At the front end, a thrust washer is interposed between the idler and the casing, and at the rear the bush in the casing is flanged to take the thrust. Further details of all the gears and pinions are given in the accompanying table.

A D.T.D.424 casting forms the top cover of the box and houses the selector rods and striker mechanism. The En 32B striker lever is a separate forging with a split boss clamped round a horizontally mounted control spindle of B.S. 5009 / 211. This spindle is 1⅛ in outside diameter by ⅜ in inside diameter. A flange is welded on its front end to receive the coupling at the end of the rod from the gear shift control lever. The shaft on which the striker lever is mounted is carried in bosses in the turret casting. It bears in each for a length of 1⅛ in, and a lip type oil seal is fitted at the front end.

The lower end of the striker lever is case hardened and the striker forks for the third, second, first and reverse gears are forged integrally on the front ends of the ¾ in diameter En 32B selector rods, while those for fourth and fifth speeds are on the selector fork bosses. A slotted plate, forming the interlock, is fitted over the end of the striker rod; its front and rear edges register in horizontal, transverse slots in the cover assembly. The slot in the plate is so shaped as to allow the

GEARBOX DATA

	Pitch circle diameter	Diametral pitch	Material	Face width of teeth
Primary gear	3·5714 in	7	En 39A	1 $\frac{5}{16}$ in
4th speed gear	4·714 in	7	En 39A	1 $\frac{5}{16}$ in
3rd speed gear	6·142 in	7	En 39A	1 $\frac{5}{16}$ in
2nd speed gear	7·428 in	7	En 39A	1 ⅛ in
1st speed gear	8·1667 in	6	En 39A	1 $\frac{5}{16}$ in
Layshaft cluster:				
Layshaft (constant mesh) gear	7·285 in	7	En 39A	1 $\frac{5}{16}$ in
4th speed gear	6·142 in	7	En 39A	1 $\frac{5}{16}$ in
3rd speed gear	4·714 in	7	En 39A	1 $\frac{5}{16}$ in
2nd speed gear	3·4285 in	7	En 39A	1⅛ in
1st speed gear	2·6667 in	6	En 39A	2⅛ in
Reverse speed gear	4·333 in	6	En 39A	$\frac{15}{16}$ in
Reverse idler gear	4·1667 in	6	En 39A	⅞ in

The spiral bevel pinion is straddle-mounted

striker to move freely over its full fore and aft travel without moving the plate. However, if the striker is moved laterally, the plate slides horizontally with it.

Two blocks are riveted to the underside of the plate on each side of the centre of the slot in it; they register between the arms of the striker forks when in the neutral position. The distance between the two blocks is such that two striker forks cannot pass between them simultaneously; thus, it is only possible to select one gear at a time. A baulk for first and reverse speeds is incorporated in the casting that carries the gear shift lever. To engage these speeds, it is necessary to lift the lever to clear the catch-plate in the bottom of the casting.

A split boss on the En 16B or C selector fork for second and third speeds is clamped on the rod by a $\frac{1}{2}$ in diameter pinch-bolt. The fork for fourth and fifth speeds is also of En 16B or C, but it is retained by a conical ended set screw in a $\frac{3}{8}$ in diameter B.S.F. tapped hole in its boss. The conical end of this set screw registers in a tapered hole in the rod. For first and reverse speeds, a striker fork is secured by a pinch-bolt to the rear end of the rod. A peg, screwed into the selector fork and secured by a split pin, registers between the two lugs of this striker. The selector fork is pivoted at its upper end to the top cover.

A conventional spring-loaded ball type gear lock is housed in the D.T.D.424 casting that is bolted to the top cover to support the rear ends of the rods. A similar casting supports the front ends of the rods. Separate, bolted-on castings are employed instead of carrying the rods in bosses cast integrally in the cover so that, in the event of wear being experienced after extended service, replacement can be effected inexpensively. Moreover, this also saves a line-boring operation during manufacture.

Rear axle

A Hardy Spicer 1600 propeller shaft transmits the drive to the rear axle. It has needle roller universal joints at its ends and the tube is 3 in diameter × 0·120 in thick × 32¼ in long. The sliding joint is at the front end. In the laden position, the axis of the shaft slopes down towards the rear at an angle of 1 deg from the horizontal.

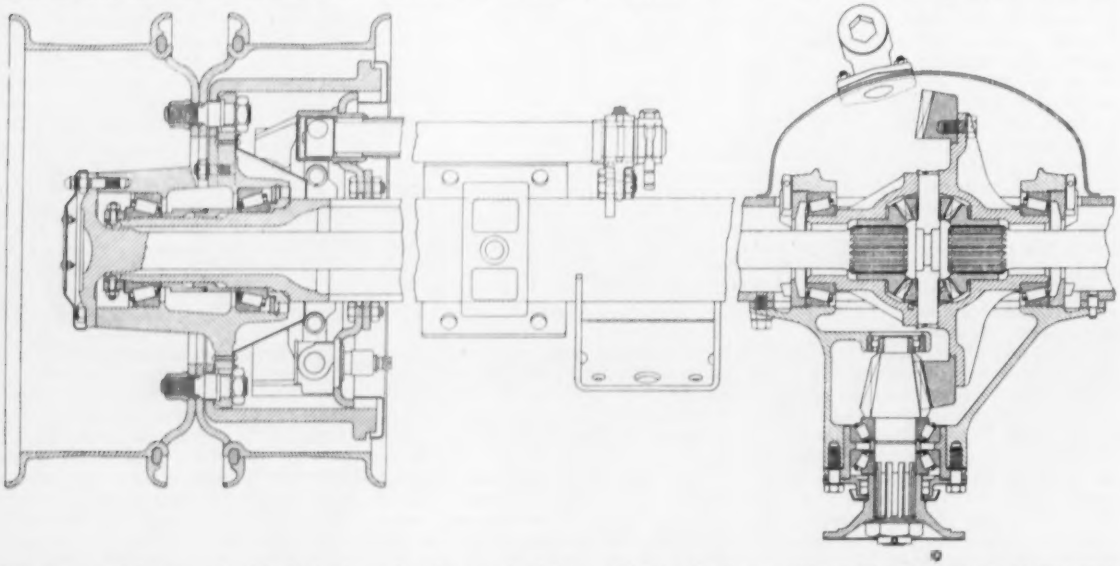
A single reduction spiral bevel final drive is employed. It is housed in a banjo type casing formed by welding together two steel pressings; the half shafts are fully floating. Both the gearbox and the axle are installed with the axes of the mainshaft and spiral bevel pinion at an angle of 1 deg from the horizontal, so that, as viewed from the side, the transmission line is straight. However, in plan view, the propeller shaft is set at an angle of 1½ deg to the longitudinal centre line of the vehicle. There are four alternative final drive ratios to suit different

A malleable cast iron gear carrier is employed

customers' requirements. They are: 4·7:1, 5·22:1, 5·87:1, and 6·28:1; the lower two ratios are primarily intended for goods vehicles. The oil capacity is 2½ gallons and S.A.E. grade 90 is recommended.

The gear carrier is a B.S.310 grade 3 malleable iron casting. It is secured to the banjo casing by twelve ½ in diameter studs. The axis of the spiral bevel pinion is offset 1½ in to the right-hand side of the centre of the differential pinion spider. There are 10, 9, 8 or 7 teeth on the pinion, according to the ratio, and the back lash is 0·008-0·011 in. The pitch circle diameters of the pinions are 3·290 in, 2·9605 in, 2·6315 in and 2·456 in respectively.

An En 39A pinion is employed and its integral shaft is 2 in nominal diameter. It is carried in two opposed taper roller bearings, with their centres approximately 1$\frac{7}{16}$ in apart; additional support is afforded at the rear by a parallel roller bearing, in which is carried a 30 mm diameter spigot extension of the rear end of the pinion. A boss in the casting houses the outer



Arrangement of the rear axle. The brake drums have been made of heavy section to give them a relatively high heat capacity to keep their temperature to a minimum

race of this bearing; the inner race is of the parallel type that permits a limited amount of axial float. The assembly comprising the internally splined boss of the companion flange for the universal joint and the inner races of the two taper roller bearings, with a $\frac{3}{8}$ in long distance tube between them, is pulled against the gear by a castellated nut and plain washer on the $1\frac{1}{4}$ in B.S.F. threaded front end of the spindle.

The outer races of the taper roller bearings are in a malleable cast iron housing spigoted into the end of the nose piece. This housing is flanged and secured by eight $\frac{1}{2}$ in diameter set bolts. The outer races are inserted one from each end of the housing and are separated by a collar formed in it. A cast iron cover closes the front end and also houses a lip type oil seal that bears on the boss of the companion flange for the universal joint. The seal housing is enshrouded by a dished thrower ring welded on the companion flange. Adjustment to the pre-load on the taper roller bearings is effected by shims between the inner race of the front bearing and the distance tube. Shims are fitted between the flange round the bearing housing and the end of the nose piece for adjusting the axial position of the pinion. The thickness of the shims needed is predetermined by gauging the carrier and the pinion assembly.

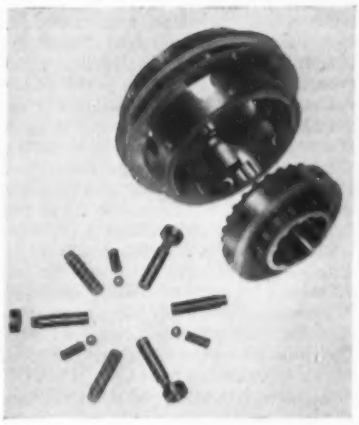
An En 39A crown wheel is employed; it is $15\frac{1}{2}$ in overall diameter and has 47 teeth for the three highest ratios and 44 for the lowest one. It is spigoted on to the flange round the differential cage and is secured by twelve $\frac{5}{16}$ in diameter set bolts, which are passed through clearance holes in the flange and screwed into it. Behind the flange, substantial radial ribs between the bolts stiffen the component.

The cage is a two-piece casting of B.S.592 grade 1 iron. An unusual feature is that the two pieces are spigoted together. The arms of the En 34 spider do not have any flats machined on them. Holes drilled between the teeth of the En 32T pinions ensure that there is a continuous flow of oil through the Clevite bearing bushes in the bores of the pinions. These bushes are $1\frac{7}{16}$ in long.

Another unusual feature of the assembly is that the 2 in diameter phosphor bronze thrust washers between the ends of the pinions and the cage are flat on one side, where they bear against the pinions, and of spherical form on the other, where they seat in the cage. They are each located against rotation by a shouldered dowel. The larger diameter portion of each dowel is in the cage and the small one projects into a hole in the washer. This hole in the washer is large enough to allow the washer to align itself correctly, but not so large as to allow the dowel to pass right through it and score the end of the gear. The provision of these washers is an example of the detailed attention that has been devoted to reduction of servicing costs.

Synchronizer and dog clutch assemblies for second and third speeds

The En 36T differential gears are 5 in overall diameter, and the effective length of tooth engagement is $1\frac{7}{16}$ in. Their back lash is 0·007-0·009 in. Plain phosphor bronze thrust washers,



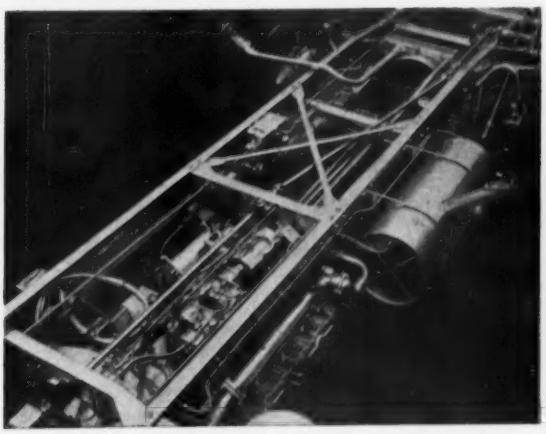
Synchronizer and dog clutch assemblies for fourth and fifth speeds

$4\frac{5}{32}$ in outside diameter by $3\frac{1}{8}$ in inside diameter, are interposed between the outer faces of the gears and the cage. The gear bosses are 3 in diameter and they bear for a length of $1\frac{5}{16}$ in in the bore in the cage. Their diametral clearance in the cage is 0·008 in.

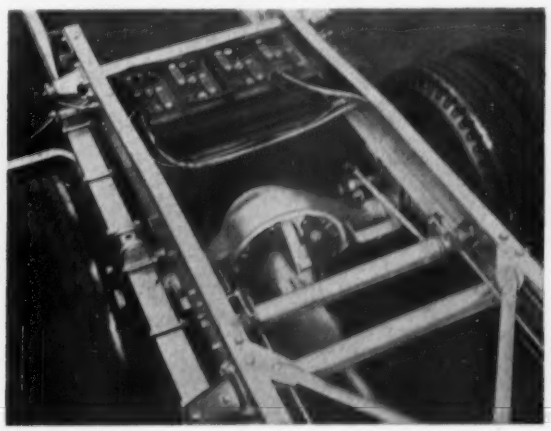
On each side, the cage is machined and ground to about $3\frac{1}{4}$ in diameter for the taper roller bearings that carry the assembly. These bearings are spaced with their centres about $10\frac{7}{8}$ in apart. They are housed in eyes in the gear carrier. Both the pre-load of the bearings and the mesh of the crown wheel is adjusted by ring nuts screwed into the outer ends of the eyes to bear against the outer races. The inner races, of course, are pulled against shoulders round the cage.

The half shafts are of En 26. They are 2·3 in diameter over the splines at their inner ends and about 1·81 in diameter over the length between the splines and their upset driving flanges. The spline root diameter is 53 mm and the depth is 2·75 mm. The En 14 axle tube is 5 in diameter and $\frac{1}{4}$ in thick at its ends.

The wheel hub is a B.S.310 grade 3 malleable iron casting. Two opposed



The engine installation and front end of a chassis with the Clayton Dewandre vacuum servo unit installed



At the rear end of the chassis the brake controls are connected to a cross tube

taper roller bearings are assembled into it from each end; their centres are approximately 5½ in apart. The inner race of the outer bearing is on a 3½ in diameter portion of the axle, and the diameter at the inner bearing is 3⅝ in. A two-piece distance sleeve separates the two inner races. Each sleeve is shouldered externally at one end and internally at the other so that one spigots into the other. Shims are fitted round the spigot joint to allow the axial float of the bearings to be adjusted to approximately 0·010 in. This arrangement has been adopted so that the shims are not damaged when the hub assembly is fitted on to or removed from the axle.

A ring nut, drilled radially to accommodate two locking bolts, is screwed on the 3 in diameter threaded end of the stub axle and pulls the whole assembly against a distance ring which, in turn, bears against a shoulder at the inner end of the stub axle. This ring also is shouldered and its smaller diameter portion bears against the inner race of the inner bearing. Round this portion is a mild steel ring housed in the end of the hub and retained by a snap ring in a groove in the housing. The function of this mild steel ring is to ensure that when the hub is removed, it comes off complete with its bearing assembly.

A lip type seal is pressed into the housing after the snap ring has been placed in position and bears round the larger diameter portion of the distance ring. Should any lubricant escape past the seal, it is trapped in a cupped steel shroud, which is fitted over the inner end of the hub and riveted to the flange for the wheel and brake drum. Holes are drilled through the flange to allow this grease to pass out clear of the brake drum. The outer end of the hub is sealed by the driving flange.

The ten ⅞ in diameter studs that secure the twin wheels and B.S.1452 grade 17 brake drum to the flange

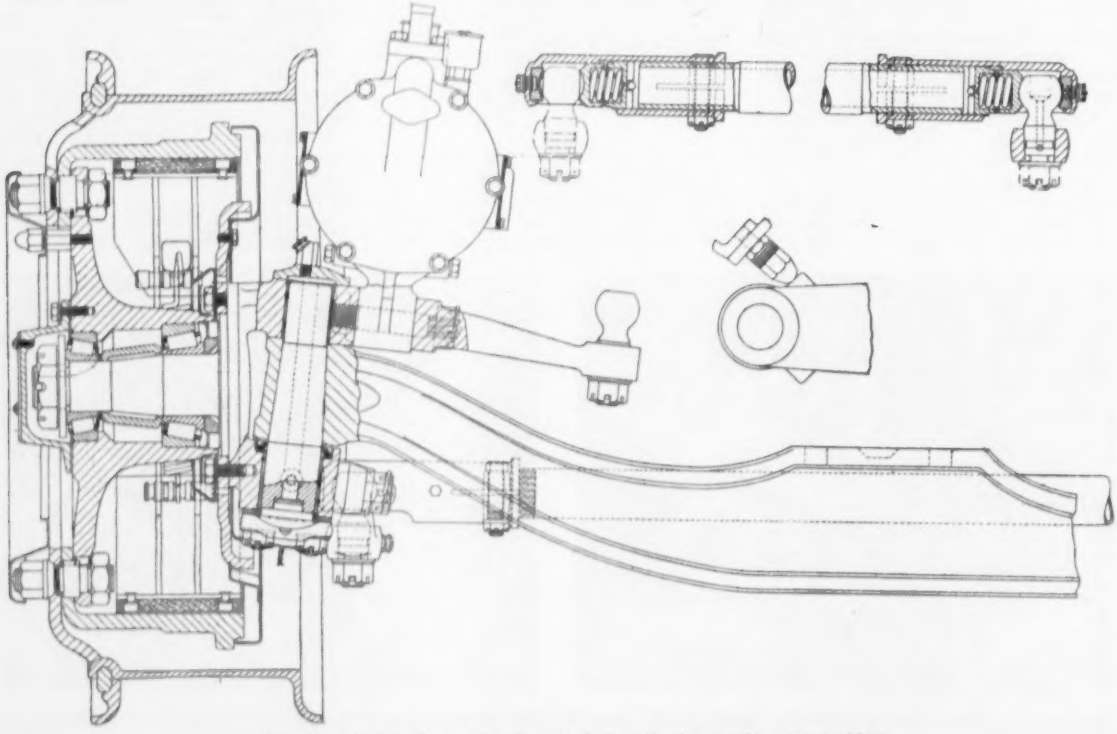
Shackle stops are fitted to avoid serious trouble in the event of failure of the top leaves of the springs

round the hub have a collar formed round them near their centres. They are threaded at each end. One end of each stud is passed through a hole in the flange, and a nut is assembled on to it and tightened to pull the collar into a counterbore in the outer face of the flange. Each stud is located against rotation by a peg in a ¼ in diameter hole drilled through the collar and into the flange. The brake drum is carried on a 1¼ in diameter portion of each stud. A conical face between the 1¼ in diameter portion and the ⅞ in diameter threaded portion of each stud forms the seating for the inner wheel, and the nuts have conical faces that seat in countersinks in the holes in the outer wheels.

Rear suspension

Semi-elliptic springs are fitted at the rear. Rubber bump stops are mounted under the frame side members to bear directly on the axle tube. The spring rate is 0·351 ton/in in the bus chassis and 0·440 ton/in for the coach. From the free camber to the fully laden positions the deflections are 6·42 in and 6·25 in respectively and from the free camber to the full bump positions they are 8·4 in and 8·5 in respectively. The unsprung weight is 1,810 lb.

The En 45 spring is 60 in long between the eye centres. On the bus it has sixteen leaves ⅜ in thick and one leaf ½ in thick, whereas on the coach twenty ⅜ in leaves are fitted with one ½ in thick leaf. The second leaf on each spring is extended round the spring eye in order to prevent complete collapse in the event of breakage of the top leaf. Shackle stops are also employed as an additional precaution. All the leaves are 3½ in wide. The bushes are of phosphor bronze and the 1·060 in diameter En 32B pins are case hardened and chromium plated. Spring hanger brackets of B.S.310 grade 3 malleable iron are bolted to the outer faces of the frame side members.



At each end of the front axle, the swivel pin is firmly seated in a tapered hole

Front suspension

At the front, the reversed Elliott-type axle, with conventional semi-elliptic springs, is employed. It is of I-section En 16B or C. At the centre its overall dimensions are 3½ in deep by 2¾ in wide over the flanges and the web thickness is ¾ in. Rubber bump stops are fitted on top of the springs, and in the full bump position they bear on the underside of the frame side members. Armstrong AT11 double-acting dampers are employed. The swivel pin angle is 5 deg, the castor angle 1½ deg and the camber angle is also 1½ deg. There is no toe in.

The spring rate is 0·347 ton/in on the right-hand side and 0·301 ton/in on the left-hand side. This difference, of course, is to allow for the asymmetric weight distribution due to the engine being mounted with its centre of gravity to the right of the centre line of the vehicle. On the right-hand side, the deflection from free camber to the laden position is 4·47 in and to the full bump position it is 6·6 in, whereas on the left-hand side the deflections are 4·65 in and 6·8 in respectively. The material and dimensions of the springs are the same as those at the rear except that they are 56 in long between eye centres, and on the left-hand side ten ⅛ in thick leaves are fitted, while on the right-hand side there are twelve. As at the rear, the bushes are of phosphor bronze and the 1·060 in diameter, En 32B pins are case hardened and chromium plated to give long life before replacement is necessary. The spring hanger brackets are of B.S.310 grade 3 and they are mounted under the frame side members.

Another example of the careful attention devoted to detail to give the vehicle a long life between overhauls is the case hardening and chromium plating of the En 34 swivel pins. The two treatments are used because chromium plating not only takes better on a case hardened surface but also lasts longer than when it is on an unhardened surface.

The lower bearing of the swivel pin is 1·890 in inside diameter × 1¾ in long, and the upper one is about 1·5 in diameter × 2 in long. Both are Glacier, lead bronze lined, steel bushes. The centres of the bearings are spaced about 5½ in apart.

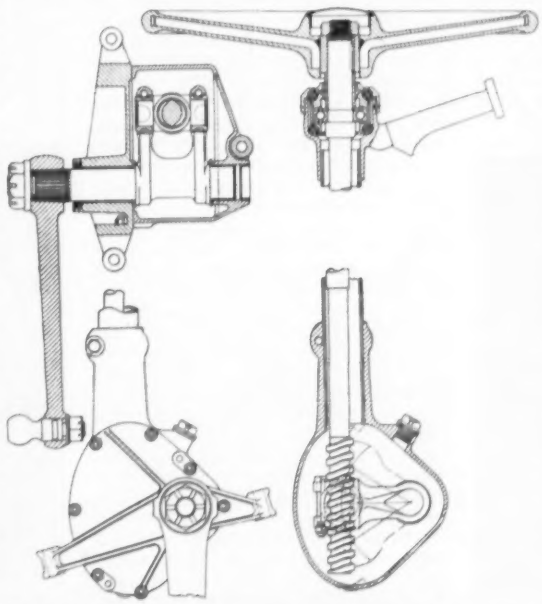
Between the two bearings the pin is tapered, where it is carried in the boss at the end of the axle. The thrust between the hub and the pin and axle assembly is taken by a special bearing at the lower end. This bearing comprises a mild steel thrust pad spigoted into the lower end of the swivel pin and a phosphor bronze washer that seats in a cap bolted up to the bottom end of the swivel pin bearing housing. The phosphor bronze thrust washer is flat on its upper face, where it bears against a mild steel pad. Its lower face, which seats in the retainer cap, is of spherical form. Shims are fitted between the cap that retains the thrust bearing and the lower swivel pin bearing housing to adjust the axial location of the swivel pin bearing assembly. A lip type oil seal is fitted in the upper end of the lower bearing housing, and the upper end of the top bearing of the swivel pin is closed by another bolted-on cap.

Steering and track rod arm arrangement

A stamped En 16B or C stub axle is employed. The steering arm is also of En 16B or C and has an effective length of 6⅛ in. It is secured to the axle by two ¾ in diameter studs with slotted nuts. An unusual feature of both the steering arm and the 7 in long, En 16B track rod arm is that they are polished all over to eliminate stress raisers and to facilitate inspection. The track rod arm is pulled against a machined face on the lower knuckle of the swivel pin joint by slotted nuts on two ¾ in diameter studs.

At the inner wheel-bearing, the stub axle is 2½ in diameter and at the outer bearing it is 1¾ in diameter. Both bearings are of the taper roller type; they are separated by a barrel-shaped, cast iron distance piece. Their centres are spaced approximately 3·57 in apart. The bearings are assembled into the En 3A hub from each end. A slotted nut with a ¼ in thick plain washer is screwed on to the 1½ in diameter threaded end of the stub axle to pull the inner races and distance piece assembly against a distance ring that is shaped so as to clear the root fillet between the stub axle and the brake back plate carrier flange. A felt oil seal is fitted in the inner end of the hub and bears against the periphery of the distance ring. The outer end of the hub is sealed by a bolted-on cap. Adjustment to give the running clearance in the bearings is effected by interposing shims between the distance piece and the inner race of the outer bearing until the rollers are just nipped; then an additional 0·010 in thick shim is added to give the correct clearance.

The wheel and brake assembly is conventional. As at the rear, the S.M.M.T. standard 10-stud fixing arrangement has been adopted. All the studs are ⅞ in diameter with B.S.F. threads. Wheel-nut guard rings are fitted. The wheels are of the three-piece, welded type with 20 in diameter rims. The rim widths are 6·00 in when 9·00—20 tyres are fitted and 7·33 in when 10·00—20 tyres are employed, and the wheels are offset 5·375 in and 6·05 in respectively.



The bearing that carries the upper end of the steering column is flexibly mounted

Steering

A.E.C. worm and nut type steering gear is employed. The worm, which meshes with a bronze nut that is trunnion-mounted, is formed at the lower end of the steering shaft. The two trunnions are carried on a short crank formed on the rocker shaft. Worm shaft thrust is taken by a self-aligning ball bearing at the upper end. The bearing is mounted between rubber rings, which absorb shock and allow for the slight rocking action of the steering shaft. This rocking action, of course, is due to the fact that the bronze nut moves through an arc instead of in a straight line. Oil bath lubrication is furnished for the worm and nut. A lip type oil seal at the lower end of the shaft prevents oil seepage out of the unit.

The steering wheel is 20 in diameter and is secured by means of a nut on the end of the steering shaft. A clip-on

cover conceals this attachment. The wheel is of coloured Vulcanite compound, moulded over a built-up steel frame.

A conventional drag link and one-piece track rod layout has been adopted. The drop arm is of En 16B or C. It is 12⅛ in long and is polished all over. The drag link, or steering rod, is 1¼ in outside diameter × ¼ in thick and is of B.S.5009-211 steel tube. Spring-loaded ball joints are fitted at the ends, and the centre-to-centre length between them is 55 in. The cross sectional dimensions of the track rod are the same as those of the steering rod and it also is of B.S.5009-211 tube. Its centre-to-centre length between the spring-loaded ball joints is 68 in. The steering ratio is 32:1 in the mid position. This gives 5½ turns from lock to lock and the turning circle is 62 ft diameter. The wheel angles at the full lock positions are 41¼ deg and 33½ deg.

Brakes

Girling leading and trailing shoe type brakes are employed. On the right-hand control chassis the Clayton Dewandre triple vacuum servo control system can be used, while the air pressure system can be fitted to left- or right-hand control versions. The drum diameter is 15½ in. All the shoes are ½ in thick and their widths are 4½ in at the front and 6 in at the rear. This gives a total friction lining area of 256 in² at the front and 362 in² at the rear. The drum flanges are of heavy cross section, about ¾ in thick, so that they have a large heat capacity and therefore their temperatures remain low during operation.

The brake pedal is carried on a phosphor bronze bush, 1⅛ in outside diameter × 1¼ in long, on an En 32B cross shaft. This shaft is common to both the brake and clutch pedals; it is case hardened at the bearing surfaces. The lever ratio of the brake pedal is 3·25:1, and the linkage increases the ratio to 3·75:1. This gives a pedal travel of 7¼ in.

Air pressure brake actuating cylinder arrangement

When the vacuum servo braking system is installed, the arrangement is as follows. A short link connects the lower end of the brake pedal with a bell crank lever pivoted on a bracket at the front end of the longitudinal member that carries the cross shaft. The other end of this lever is connected to another bell crank lever carried by a bracket on the front cross member at a point about 5 in inboard of the side member on the near-side of the frame. From this bell crank there is a straight control-run back through the servo unit to an upstanding lever on a tubular cross shaft about 17 in in front of the back axle.

Pendant levers near each end of this cross shaft are connected by tie rods

The Clayton Dewandre reservoir and valve unit of the Bendix Westinghouse pressure braking system

to levers on the brake camshafts. The camshafts are supported in the usual manner in the brake back plate, and also in phosphor bronze bushes in bosses near their inner ends. These bosses are each secured by 2 bolts to a ⅞ in thick plate welded to the axle tube. There is sufficient clearance between the bolts and their holes in this plate to allow the camshaft to be correctly aligned during assembly.

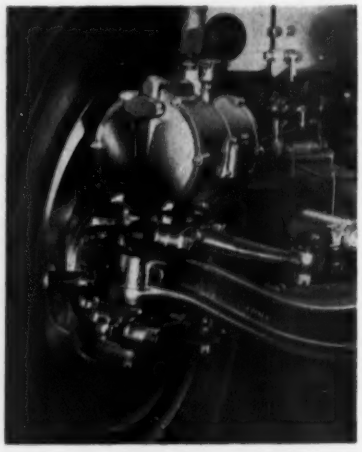
The main servo motor is the Clayton Dewandre VH1/812L type and is mounted on the inner face of the near-side frame side member. Auxiliary motors, to operate the front brakes, are mounted one on each front axle swivel pin knuckle. Each motor is of the tandem cylinder type, and the degree of vacuum in the two cylinders is regulated by the main servo valve. Synthetic rubber dust covers are fitted on all servo motor piston rods.

A vacuum tank, of 4,200 in³ capacity, is mounted on the outer face of the frame side member alongside the main servo. In this tank, a vacuum of 22-24 in of mercury is maintained by a Clayton Dewandre exhauster, type B550. This exhauster is driven in tandem with the fuel injection pump. The degree of vacuum in the reservoir is indicated on a gauge in the driver's instrument panel.

When the air-pressure control system is installed, the brake camshafts are actuated directly by diaphragm type pistons in expansion chambers mounted on each axle. Admission of air to the brake chambers is controlled by a D1 type, pedal and valve assembly mounted on the driver's floor plate. The design of this assembly is such that, until the pressure in the brake line reaches 75 lb/in², it is proportional to the force applied to the pedal. After this, the full pressure in the reservoir, approximately 100 lb/in², is released into the brake lines. This enables a powerful brake application to be made in an emergency.

One of two types of air compressor is fitted. The choice is between the Clayton Dewandre, type PCGA 198 unit, with a bore and stroke of 2¼ in and 1¾ in respectively and a piston displacement of 7·2 ft³ per minute at 1,000 r.p.m. of the compressor, and a Westinghouse, type E7 unit, with a bore and stroke of 2¼ in and 1¾ in respectively, and a displacement of 8·0 ft³ per minute. Both are twin cylinder, air-cooled units, and each is mounted in tandem with the fuel injection pump. The installation is supplied with oil from the engine pressure-lubrication system. Air is drawn through the engine air cleaner and is discharged into a single-chamber reservoir on which are mounted a safety valve and an unloader valve. The reservoir pressure is indicated by a gauge on the driver's instrument panel.

The hand brake control is of the single-acting lever type with a ratio of 8·41:1. This ratio is increased to 14·7:1 by the linkage. The control lever is mounted on the outer end of a cross shaft supported in two phosphor bronze bearings in housings, one bolted on the longitudinal frame member supporting the driver's foot plate and the other on the frame side member. A pendant lever on the inner end of the cross shaft is connected to a control rod that extends back to the cross shaft, which is immediately in front of the rear axle. With the vacuum actuation system, the foot brake linkage is also attached to this cross shaft. This control-run is about 1 in



The mounting arrangement for the Clayton Dewandre servo unit is the same as that for the air pressure unit

inboard of the offside, frame side member.

Frame

In general, the frame is of conventional form. Its overall height from the ground when 9·00—20 tyres are employed, is 2 ft 9½ in at the centre. The front and rear overhangs are 5 ft 1⅛ in and 8 ft 1½ in respectively. An unusual feature is the cross bracing in the bays immediately in front of and behind the engine. The cross bracing has been incorporated to give increased resistance to lozenging and whip. This arrangement is a good feature in that it gives adequate stiffness without adding appreciably to the weight of the frame: the alternative would be to increase the gauge of the main members of the structure. During assembly, the end plates of the cross bracing ties are first bolted to the flanges of the frame side members and then the ties are heated and welded on. As they cool and contract, they apply a pre-load to the bay. They are also welded together at their centres to prevent chafing and rattling.

At the centre, the channel section side members are 8 in deep × 2½ in over the flanges × ¼ in thick for the home market, and 8⅛ in deep × 2½ in over the flanges × ⅜ in thick for export. They taper to a depth of 6½ in at the front and rear ends. The overall width of the frame over the side members is 2 ft 4⅜ in.

There are nine main cross members and, as has already been stated, they are cross braced with mild steel strip, 1½ in wide × ⅛ in thick. At the front a 15-20 ton carbon steel, cold formed channel is riveted on. It is 6½ in deep × 2 in wide × ⅛ in thick. The second and third cross members are fitted between the front and rear spring hanger brackets. They are of B.S.980/ERW2, 3 in outside diameter × 8 gauge tube.

A fourth cross member is 6 ft 3½ in behind the front axle, and it supports the front end of the engine. It is 28-33 ton carbon steel channel section, ⅜ in thick × 5 in deep and 2½ in wide at the centre. To support the rear of the engine there is another cross member, 4 ft 3 in behind the one that carries the front of the engine. This member is ⅜ in thick, 28-33 ton carbon steel channel section, 7⅛ in deep and 2½ in wide at the centre. Two more tubular members are fitted, one between each pair of rear spring mounting brackets. Their cross sectional dimensions are the same as those at the front. Alternative rear extensions of the frame can be fitted for buses or coaches. One is horizontal and the other is cranked downwards. These extensions are both 4 ft 8 in long.

At the front, the driver's platform is on an outrigger frame mounted on the side member. An L-section 26-32 ton carbon steel, longitudinal member, 36 in long × 3⅞ in wide × ⅜ in thick, forms the main component of this frame. At its front end, it is bolted to the front cross member, and at the rear to a tapered channel section outrigger of 28-33 ton carbon steel.

Electrical equipment

A 24 volt battery of 135 amp-hr capacity is employed. The engine starter is a Simms 524SGR 107/4, 5 in diameter axial engagement type and the dynamo is a Simms 6 in type 624 YR2DV for buses or the 7 in type 724 DQR20C/5 for coaches. It is driven at 1·86 times engine speed. The maximum output of the 7 in diameter unit is 1,390 watts and its maximum load is 43 amps. The output of the 6 in diameter unit is 840 watts and the maximum load 30 amps. Simms LB11/1 and 6401 head and side lamps are fitted. The main fuses are of the 120 amp type and there are three 10 amp fuses for the lights. Two 20 amp fuses protect the auxiliaries. Lucas WT 614 Windtone or Clearhooter CT750 horns are supplied.

Other features

The radiator is mounted ahead of the front axle beneath the saloon floor. It is carried on rubber mountings on brackets bolted to the frame side members. Longitudinal movement is prevented by an adjustable tie rod, with rubber bushed end fittings, secured to a frame cross member in front of the radiator.

In the block, vertical tubes with horizontal gills form the core. This assembly is bolted to separate top and bottom tanks and side pillars of cast aluminium. In chassis for the home market, four rows of tubes are fitted; this is adequate for ambient temperatures of up to 85 deg F. On the other hand, export chassis have six rows of tubes. The frontal area of the core block is 2·66 ft², and the capacity of the system is 7 gallons.

An air release pipe is led from the top tank into the filler pipe. A pressure relief valve and a drain are incorporated in the filler pipe, which is connected to the filler neck by a short rubber hose. The relief valve setting is 3½-4½ lb/in². A bright chromium plated, quick-release type of filler cap is fitted.

The 20 in diameter, high speed fan is driven from the engine crankshaft by a two-piece shaft. A Layrub universal joint is used at the rear end of the shaft, which is supported by an intermediate bearing. The fan is carried on a 1 in × 2½ in × ¾ in self-aligning ball bearing.

A 25-gallon, welded steel fuel tank is mounted on the left-hand side member of the frame, forward of the rear axle. It is equipped with a Simms magnetic fuel gauge and a quick-release, captive filler cap of polished aluminium. For the home market, a C.A.V., single paper element filter is incorporated in the fuel system between the injector pump and the lift pump. On export models, a bowl type sediment trap for primary filtration is fitted between the supply tank and the lift pump. In addition, twin C.A.V. paper element filters, with a parallel feed head, are incorporated between the fuel lift pump and the injection pump.

The straight-through type silencer is suspended from the right-hand side member of the frame. It is connected by a flexible metallic hose to the engine exhaust manifold. From the silencer outlet, a tail pipe is taken beneath the rear axle to a point behind the wheels; it discharges on the same side of the vehicle as is the control platform. The nominal outside diameter of the exhaust pipe throughout the whole of its run is 2½ in.

CRANKSHAFT FATIGUE

THE Motor Industry Research Association has carried out a number of tests to determine the amount by which the bending fatigue strength of crankshafts can be improved by cold rolling. The crankshaft fillets were rolled by steel balls under heavy load. Production crankshafts of forged steel and graphitic cast steel were used in the tests that are the subject of the most recently published report, No. 1954/2, which can be obtained, at a price of 10s., from the Association at Nuneaton, Nr. Lindley. This report is entitled "The Improvement in the Bending Fatigue Strength of Producing Crankshafts by Cold Rolling."

Rolling the fillets increased the strength of the forged steel crankshafts under reversed bending loads by about 60 per cent, and under one-way loads by nearly 80 per cent. With the graphitic cast steel crankshafts, which were tested only in reversed bending, scatter of results prevented close determination of fatigue limits, but an estimated value for improvement of strength of at least 60 per cent appeared to be reasonable.

Although minute cracks sometimes occurred in the rolled fillets of the forged steel shafts at loads less than those that caused breakage, they did not appear to be detrimental. In any case, the loads at which they occurred were much higher than the loads producing failure in unrolled specimens.

Henry Meadows Ltd.

IN the May 1955 issue of this journal there was a transposition of illustrations in the article "Meadows Diesel Engines." The performance curves on p. 203 are those for the six-cylinder engine, while those on p. 205 are for the four-cylinder engine.

HARD-FACED TOOLS

The Application of Stellite Alloys to Extend Working Life

IN either continuous or long-run production the maintenance of tools is an ever-present problem. Particularly in the hot working of metals, where the rate of wear may be relatively high, the reconditioning or replacement of tools may itself have to be put on a regular production basis. Substantial economies are being effected by protecting new tools, or repairing worn or damaged ones, by facings, inserts or tips of Stellite alloys, which are highly resistant to heat and abrasion. A repaired tool can have a production life exceeding that of a new steel tool and, furthermore, repairs can be repeated a number of times.

Of the Stellite alloys, two are particularly suitable for protecting hot-working tools. Stellite 6 is a cobalt-base alloy having a hardness, when deposited, of 375-450 D.P.N. (Rockwell C 38-45). It softens but little at red heat and has exceptional resistance to abrasion by hot scaled steel. It is machinable only with carbide tools but is easily ground. Alloy "C" is a nickel-base alloy which, when deposited, has a hardness of 250-300 D.P.N. (Rockwell C 24-30) and can be machined with tools of high speed steel. It possesses excellent resistance to heat and to impact. In service it rapidly work-hardens by impact and will then resist abrasion satisfactorily. Both are supplied as welding rods of $\frac{1}{8}$ in, $\frac{5}{32}$ in, $\frac{3}{16}$ in, or $\frac{1}{4}$ in diameter, either coated for arc deposition or bare for other electric methods (atomic hydrogen, Heliarc, argon arc) or oxy-acetylene flame.

Selection of the more suitable grade of alloy for specific tools can best be indicated by consideration of the various aspects of forging practice. For hammer dies, Alloy "C" is commonly used as it hardens rapidly under impact. By virtue of its relative softness in the deposited condition, either new or worn tools can be faced with this material and then machined by the normal methods with H.S.S. tools as used for steel. The working life obtained will at least equal and will usually exceed that obtained with tool steel facings or inserts. If the form can be produced by grinding, or by machining with carbide tools, even better results can be obtained by the use of Stellite 6.

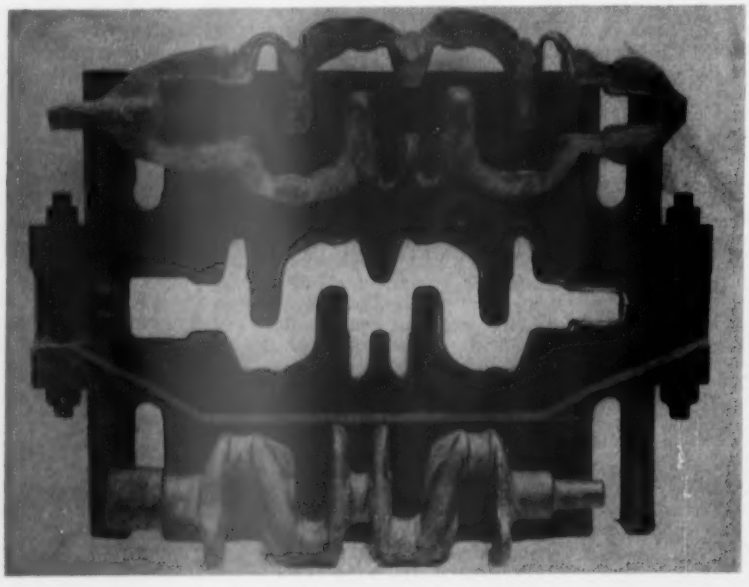
PHYSICAL PROPERTIES
Deloro Stellite Alloy "C"

Hardness	250-300 D.P.N.
Ultimate tensile strength	30 ton/in²
Yield point	20 ton/in²
Elongation (on 2 in)	10-15 per cent

Since impact is virtually absent in press forging operations, Alloy "C" is not suitable for upsetting or piercing punches, grip dies or die inserts. For such applications, Stellite 6 is used.

Hot clipping beds and punches may be faced with either of the two alloys. Where an irregular shape presents difficulty in finishing, Alloy "C" is used and finished by machining and filing. Where it is practical to grind to finish shape, however, Stellite 6 is preferred, as it will maintain a sharp edge for a longer run before a face regrind becomes necessary. This characteristic is of particular value when the tool is required to take relatively deep cuts on flash that is thicker than usual as a result of wear on the forging dies. Stellite 6 is also used for cold-clipping.

Blades for cropping hot forgings are also faced with either alloy and the selection will be made to suit specific production requirements and capacity. Stellite 6 will, of course, ensure less interruption of production for re-grinding.



Crankshaft trimming die faced with Stellite 6

Tool preparation

In forging dies the wear is likely to be more pronounced in particular areas, and it may be sufficient to protect only these areas, instead of completely facing the impression with the hard alloy. Accordingly, one or more recesses are cut in the tool and filled with the alloy, which is then finished flush with the original contour. Any convenient method, hand or machine, may be employed for recessing, but certain stipulated provisions should be observed to ensure satisfactory results.

The depth of the recess should provide a minimum thickness of the finished deposit of $\frac{1}{8}$ in and a maximum thickness, in the case of Stellite 6, of $\frac{1}{4}$ in. Alloy "C" can be used to a greater depth. Where small or intricate shapes are required, a cavity of suitable size should be machined and filled with the alloy. The impression is then sunk directly into the alloy.

Sharp corners in the recess should be avoided and a $\frac{1}{4}$ in radius or chamfer is desirable both to simplify the actual deposition of the alloy and to obviate the possibility of an unsound result. A recess should be so designed that there are no feather edges to the deposit when it is finish machined or ground. Minimum dimensions for recesses in shear blades are given in the accompanying sketch.

When repairing worn or damaged tools it is important to remove completely any cracks. If to do this entails relatively deep cavities, they may be filled with Alloy "C" with, if necessary, a further deposit of Stellite 6.

Forging dies for thrust plates. Faced with Alloy "C"

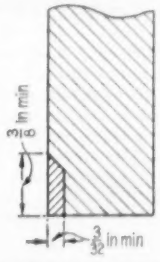
Methods of alloy deposition

While the choice of method will be limited by the equipment available, deposition can be effected in a variety of ways. Metallic arc welding gives a rapid rate of deposition with a minimum of distortion. A sound deposit with a smooth surface is obtained with the D.C. arc with the electrode positive, but A.C. may also be used. Recommended current intensities for $\frac{1}{8}$ in, $\frac{5}{32}$ in, $\frac{3}{16}$ in and $\frac{1}{4}$ in rods are 100, 130, 155 and 210 amp D.C. and 125, 155, 190 and 250 amp A.C. respectively.

When the deposition is made electrically, the arc should not be allowed to melt the steel deeply or excessive dilution of the alloy with the base will occur and, as a consequence, the wear-resisting properties of the deposit will be lowered. Each welding rod should be dried out by shorting it to earth for a few moments and the deposit should be made in a straight bead without weaving. Where several superimposed layers of the alloy need to be deposited it is essential to remove all welding slag from each layer by chipping or by scratch brushing. Any neglect in this respect may result in an unsound deposit.

After deposition has been completed the tool should be allowed to cool slowly in an insulating powder, such as vermiculite, kieselguhr, or lime. Rapid cooling may lead to thermal stresses being set up in the work and the formation of cracks in the deposit and the tool. It is advisable to deposit only sufficient alloy to finish cleanly to size, and not to build up deposits excessively.

Somewhat greater control over the shape and soundness of the deposit can be obtained with the atomic hydrogen process because a separate filler rod is used, but the rate of deposition is slower. Similar results are given with the Heliarc, argon arc, and carbon arc processes. A harder deposit is obtained by use of the oxy-acetylene torch, as there is little dilution with the base metal. This method is slower than the metallic arc but yields a sound deposit. As the heat input is greater the possibility of distortion occurring is correspondingly increased.



Recommended finished dimensions of facing for shear blades

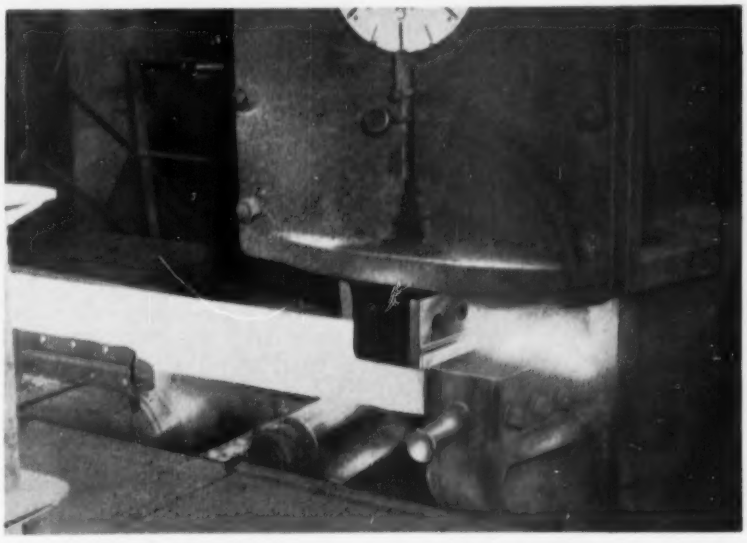
The technique of depositing the alloys by oxy-acetylene torch follows that widely used in facing engine valves and valve seatings. It is important to preheat the tool. The temperature should be raised uniformly to 300-350 deg C before commencing to deposit the alloy and this temperature should be maintained throughout the operation. This precaution reduces the risk of crack formation, particularly in the case of die steels containing tungsten. It also facilitates subsequent machining as it prevents hardening at the junction between the alloy and the steel base.

Finishing the deposit

When the tool has cooled, Alloy "C" is machinable with H.S.S. tools or, if the shape is complex, can be hand filed. Grinding by hand tool or machine presents no difficulty. Should a flaw in the deposit be disclosed during finishing, it may be cleared and "spotted" with the arc before machining to final size. In the case of repairs to faced tools, the alloy will have become work-hardened in service and any machining necessary to prepare the work for a new deposit may have to be performed with tungsten carbide tools or by grinding.

Deposits on shear blades or cropping blades are surface ground to produce a sharp cutting edge. In service it is important that the working clearance between these blades is held to a minimum to ensure the best results.

Stellite 6 is machined with tungsten carbide tools, such as Wimet N or Prolite 15A/135, at cutting speeds of approximately 30 ft/min. Grinding is best effected with normal wheels, such as A60JV or A60MV grade, at speeds of from 3,500 to 4,500 ft/min. Care must be taken to avoid overheating the work or surface cracks may be caused. If a



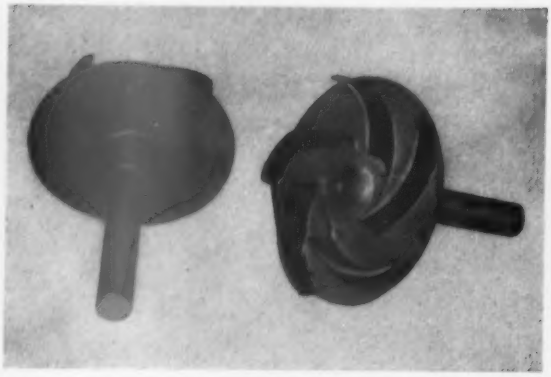
Blades of this hot shearing machine are protected with Stellite 6

coolant is used, it should be in copious supply and work should not be quenched during grinding.

Records in service

In the manufacture of engine valves in Silchrome steel, a tool in normal die steel will clip about 2,000 forgings before a regrind becomes necessary. With the die faced with Alloy "C" a typical run was 9,800 forgings.

The crankshaft trimming die illustrated, faced with Stellite 6, had clipped 9,560 forgings and was still serviceable. A standard die in tool steel begins to lose its "cut" after about 500 clippings and the edge has then to be peened out and a face regrind made.



Pump impeller cast in Alloy "C"

Another illustration shows a set of forging dies for thrust plates. The original dies in a nickel-chromium-molybdenum steel were good for a run of 1,500 forgings. When faced with Alloy "C" a run of 10,000 forgings is obtained consistently.

One producer of gear blanks, as a regular practice, obtains 3,000 forgings from a new set of steel dies. The worn or cracked impressions are then opened out, built up with Alloy "C," and machined to size. They will then produce 9,000 forgings and can be redeemed a number of times.

In the production of bolts, dies faced with Alloy "C" are reported to have produced four times the normal number of forgings obtained with steel dies and to be still retained in service. The hardened steel grips used in a bolt-pointing operation lasted for 7,000 operations. When the grips had been faced with the hard alloy, Stellite 1, they sustained 80,000 operations.

For trimming and bevelling the edges of hub cap pressings in 0·032 in mild steel, circular cutters of H.S.S. and tool steel had an average working life of three weeks. Similar cutters faced with Stellite 6 showed no wear after two months in service.

Other applications in the automobile industry include the facing of leaf-spring cropping blades, the tipping of spring coiling points and mandrels, facing circular slitting tools for sheet steel and the facing of clutch burnishing rollers.

Stellite alloys can be precision cast by the investment process, and in this form are used for drawing, coating and extrusion dies. A typical example occurs in the creel bead dies used in tyre making. To reinforce the casing beads, hoops of steel wire coated in rubber are employed. The wires, from 3 to 7 in number, enter the die through a series of parallel holes, traverse the chamber into which the rubber mix is injected under pressure, and emerge from a rectangular orifice at the opposite end of the die in the form of a flat tape of rubber with equally spaced reinforcing wires.

Prior to the use of Stellite castings, the dies were machined from high speed steel and their working life was determined by the increase in weight of the finished tape as a result of die wear. Several sets of Stellite dies have been in use more than two and a half times as long as the working life of high speed steel dies without any measurable increase in the weight of the product.

The extrusion of copper and brass rod, for machined parts of motor accessories, is carried out at temperatures up to 950 deg C and the temperature of the die may well reach 750 deg C. Since Stellite alloys soften but little at temperatures up to 1,000 deg C, and are virtually unaffected by prolonged work at that temperature, they are well suited for extrusion dies. An example of performance in this service is provided by a Stellite die for extruding a brass rod of 1·00 in nominal diameter. The first 1,000 ft produced were within the limits of 0·995 in and 0·999 in in diameter. After passing 14,000 ft the size was 0·995 in to 1·0005 in diameter, showing the amount of wear as approximately 0·001 in on the diameter. This die is still in service after extruding 33,000 ft of rod.

Alloy "C" also possesses exceptional resistance to corrosion by many common chemicals. It can thus be used, as precision-cast parts or linings, for the protection of production equipment handling corrosive substances. An illustration shows pump impellers in Alloy "C." Cast by the investment process, these require only a minimum of machining, as general dimensions can be held to ±0·005 in, critical dimensions to ±0·003 in, and a good surface finish is obtained. The Company's own foundry is equipped to produce castings to these limits.

CARBURIZED GEARS

Bending Fatigue Strength

A COMPARISON has been made of the bending fatigue strengths of the teeth of carburized spur gears finished by the following methods: (a) hobbed; (b) protuberance hobbed and shaved; (c) shaped; (d) form-wheel ground, including tooth roots; (e) ground by a generating process; (f) roots pre-formed, and teeth ground on the flanks; (g) shot peened. In addition, with one of the hobbed gears and one of the gears ground in the roots, the effect of a phosphate treatment was investigated. The tooth roots of the various gears were not all of the same form, although those produced by a number of the processes were approximately so; rather was it intended that each of the processes should be representative of good commercial practice.

Gears ground in the roots and form-wheel ground gears had markedly lower fatigue strengths than unground gears. Gears ground in the roots were also weaker than three generated ground gears that were found to have an average strength of 66 per cent of that of unground gears. The gears that had preformed roots and were ground only on the tooth flanks were not greatly affected by the grinding operation. Shot peening was beneficial, giving an improvement approaching 20 per cent. It also appears that, by using preformed roots and shot peening together, it is readily possible to produce ground gears that are, on average, about twice as strong as gears ground in the roots.

Fatigue strengths of hobbed gears are much the same as those of shaped gears of similar form. The gears which had been protuberance hobbed and shaved were slightly weaker than the hobbed gears and the shaped gears, but their fillet forms and sizes were clearly less favourable. The phosphate treatment produced an improvement of strength in the gear that had been ground in the roots.

These differences in strength, experienced with gears made by different production methods, are generally greater than differences previously shown to exist between gears made from a wide range of carburizing steels. These investigations are published in M.I.R.A. Report No. 1953/4, which is available, at a price of 10s., from the Motor Industry Research Association, Lindley, Nr. Nuneaton, Warwickshire.

BAR HEAT TREATMENT

New Plant Operated by Sanderson Brothers and Newbould Ltd.

IN order to facilitate the production of alloy steels required in the heat treated condition, Sanderson Brothers and Newbould Ltd., Newhall Road, Sheffield, have recently installed a new heat treatment plant. The plant comprises two hardening furnaces, a tempering furnace, an oil quenching unit, a water quenching unit and a revolving, floor-type charging machine. Fuel Furnaces Ltd., of Birmingham, were responsible for the design and building of the furnaces, which operate on town's gas, and for the installation equipment.

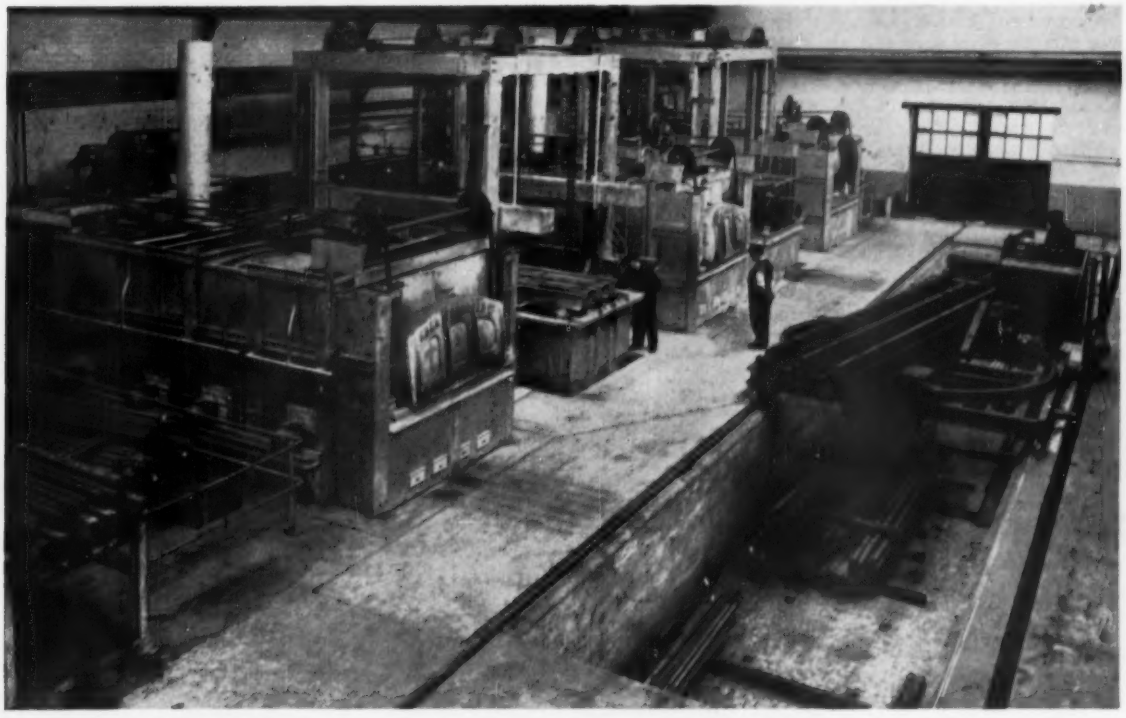
The hardening furnaces are built on the under-fired principle. Their working chamber dimensions are: width 5 ft, height 1 ft 9 in, and length 16 ft 6 in. Each furnace will take a maximum charge of three tons. In these furnaces the products of combustion are induced downwards through the load to give balanced, even heating. The proportional gas-air temperature control system is employed, each furnace being equipped with 30 burners that have measuring jets for both gas and air. By means of a recuperator in the arch and side walls, the waste heat is utilized to pre-heat the air for combustion. The hearth is specially constructed with three slots for receiving the arms of the charging machine. Compressed air is used for operating the doors.

The tempering furnace is built on the over-fired principle. It is designed for high velocity circulation of furnace gases up to a temperature of 700 deg C. For working above this temperature up to a maximum of 850 deg C, circulation is induced as in the high temperature furnaces. If desired, the re-circulating fans can be arranged to cut out automatically when the temperature reaches 700 deg C, and to re-start when the temperature falls below 700 deg C.

For normal tempering operations, which are, of course, carried out below 700 deg C, the gases of combustion re-circulate through the load and hearth at high velocity, passing through ports situated between the burner ports. Eight horizontal-type nickel chrome centrifugal fans operate in cast iron collecting chambers below the hearth. They draw the furnace gases through the load and re-inject them into the work chamber. This re-circulation of gases not only tends towards a more even temperature, but also enables the excess waste gases to be used for a frontal curtain to prevent the entry of air through the door. This greatly reduces oxidization. There are 30 burners with identical jets for gas and air. The burners are divided into two zones for temperature control purposes. An air recuperator is fitted. The working chamber dimensions for the tempering furnace are the same as those for the hardening furnaces.

As these furnaces are designed for proportional gas-air temperature control, each furnace is divided into two zones, front and rear, with a pyrometer to record the temperature in each zone. The thermocouples are inserted through the roof of each furnace. A control panel carries the six Kent Multilec temperature recorder controllers and the ancillary equipment.

Under working conditions, the reciprocating oil quenching machine has a capacity of approximately 4,300 gallons. This machine consists of a rectangular tank with cover and a substantial steel table to hold the bars. When the bars for quenching are placed on the steel table by the charger, the table, which is pneumatically operated, is allowed to sink into the tank. The cover is then automatically lowered into position. Then the quenching table, and with it the bars, is moved up and down in the tank by four pneumatic cylinders and counter-balance gear. This action is continued until quenching is completed. During quenching the oil is



Bar heat treatment department, Sanderson Brothers and Newbould Ltd.

continuously circulated by an electrically driven propeller-type circulating gear. To prevent over-heating, the oil is also pumped through a Serck cooler.

The water quenching unit is similar to the oil quenching unit in general construction, but is not provided with a cover or circulating gear. Circulation is effected by pumping cold water through the tank and thence to drain. River water, which is cheap and plentiful, is used normally, but an emergency supply of town's water is available.

There are two prefabricated steel loading tables for building up the charges of bars. One is adjacent to the hardening furnaces, the other to the tempering furnace. The construction of the tables allows for the insertion of the charging machine arms. Bars are conveyed to the tables by overhead crane.

A rotary type charging machine is arranged to work in conjunction with the furnaces, the quenching tanks and the tables for loading and unloading the bars. It is arranged to travel across the front of the installation in order to load or unload them in one movement. The machine has three arms, with a travel of 18 ft, formed from two separate horizontal members, one above the other. This arrangement allows the load to be lifted, the arms to be deposited without shock, and, finally, the arms to be withdrawn. Each of these three movements, lift, charge and travel, is driven by its own individual motor. As a safety measure, the arm travel and the rotation are interlocked so that the machine cannot be rotated unless the arms are fully withdrawn.

ROLLING MILLS

The Stanat-Mann Two-high/Four-high Combination Machine

AMERICAN designed "Stanat" combination rolling mills are now being manufactured under licence in this country by Albert Mann's Engineering Co. Ltd., Basildon Industrial Estate, Essex. The equipment will be known as "Stanat-Mann" two-high/four-high combination rolling mills, and the 1½ in and 5 in × 8 in and the 2½ in and 8 in × 8 in sizes are now available.

This new heavy-duty machine, which is designed to accomplish the work of two or more mills, can be used for bar, rod and strip reduction. In one compact machine it is possible to perform the following functions: (a) hot flat rolling; (b) cold flat break-down; (c) cold flat finishing on either a two-high or a four-high set-up; and (d) hot or cold rolling of wires and shapes.

At some time or another almost every user of a two-high strip mill has been held up by the fact that the mill was completely screwed down although further reductions were required. Hitherto one solution has been to continue processing the strip on a four-high or cluster mill with considerably smaller work rolls backed up by large supporting rolls. This two-high/four-high combination mill has been specially designed to solve this problem. To make the change from two-high to four-high, the large rolls are separated and two smaller work rolls are inserted between them. The large rolls then become "back-up" rolls.

When the unit is functioning as a conventional two-high mill, the relatively large work roll diameter is valuable for breaking-down ingots and bars, for the reduction of high lustre strip, and for skin-passing of annealed strip. Because a larger work diameter causes greater slip between the roll surface and the work material, bright and highly burnished surface finishes can be obtained when annealed strip is being reduced. An added advantage of the larger work roll diameter is its greater circumference, which reduces wear and thus improves performance and lengthens roll life between regrinds.

In flat rolling the great advantage of the four-high set-up is that with a given roll load capacity and mill horsepower, greater reductions may be made with the smaller diameter work rolls. Furthermore, in rolling the harder metals and alloys, the smaller work roll reduces the lateral spread, and hence the tendency for the strip edges to tear or crack. The "stiffness" of the backed-up four-high mill and the ease with which its small work rolls can be "crowned" makes it possible to produce extremely accurate strip with virtually no thickness variation across the width. Moreover, the smaller work roll facilitates the production of thinner finishing gauges even in highly work-hardened materials, since the length of flattened arc of contact between the work rolls and the material decreases with the decrease in work roll diameter. By employing tungsten carbide rolls, roll loads and rolling torques can be decreased even further.

Depending upon the required range and speeds of reduction, the two-high four-high combination mill can be powered by a constant-speed, or a four-speed gear-shift A.C. drive, or a variable-speed D.C. drive. The mill motor is mounted in the unit base of the machine and drives the pinion stand through double-reduction helical gearing. The pinions drive the two large-diameter rolls through universal spindles.

Stanat-Mann combination rolling mill

In this mill the back-up drive employed eliminates one of the most serious disadvantages of the conventional four-high mill—the limitation in the torque that can be transmitted by the relatively small roll necks and spindles. Another valuable feature of the back-up drive is its ability to overcome the lateral "bowing-out" of the small work rolls when heavy reductions are being taken with high front tension.

The method of providing front and back strip tension can be varied in accordance with maximum strip pull requirements and coil build-up ratios. In the simplest arrangement on a non-reversing mill, back tension is applied by a friction brake on the pay-off reel, and variable front tension is obtained by driving the coiling reel from the main mill motor through an adjustable slip clutch. Reversing mills can be equipped with a variable-voltage D.C. drive, which gives very precise tension control. The improved reductions and strip flatness obtainable with the application of front and back tensions permits successful cold working of thin, hard metals.

To accommodate heavy separating loads, the housings of the mill are high strength castings mounted on a base of fabricated steel plate construction. The housing posts are designed with exceptionally large cross-sectional areas to obtain the extreme rigidity necessary for handling severe roll loads. Anti-friction roll neck bearings are used on the back-up and work rolls, with provision for taking both radial loads and end thrust.

Both flat and grooved rolls are available for the processing of round, square and other shapes. Hot rolling can be performed over a roll temperature range up to 1,000 degF. For heat dispersion in hot rolling, the mill can be equipped with a self-contained bearing lubricant circulating, cooling and filtering system, and with means for circulating water through the roll interior.

FRONT SUSPENSION

Brief Survey of Independent Front Suspension Systems for Cars

D. R. Hume

FIVE main factors affect the behaviour and handling characteristics of a vehicle. They are: weight distribution, front suspension, rear suspension, frame or structure stiffness, and tyre characteristics. At the present stage of development these factors are all of equal importance, and indifferent or bad design in respect of one can neutralize any advantages gained by good design of the others. For instance, it is possible to change completely the handling characteristics of a vehicle simply by fitting tyres of a different make or type. However, the author considers that ultimately vehicles will, and indeed must, reach the stage of development where, for safety and other reasons, dynamic wheel loadings will be standardized for each class of vehicle. This will enable tyre manufacturers to concentrate on developing tyres with better anti-skid and anti-noise properties and which at the same time fulfil the standard slip angle requirements.

So far as front suspension is concerned, the author feels that a higher roll centre than that commonly used at present is necessary to attain better handling characteristics. This is particularly desirable for passenger cars, because of their relatively high centre of gravity. The ride characteristics of most sports cars could be improved by raising their roll centres, since this measure would enable softer springs to be used. Moreover, the feel and attitude of the vehicle when cornering would be better.

It is desirable when traversing road irregularities, for the wheels to deflect with a purely vertical motion, and they should not be subjected to interference from inaccurately or compromisingly positioned linkage. Further, the instantaneous static roll centre should be somewhere between 70 per cent and 90 per cent of the tyre rolling radius above ground level, according to the height of the centre of gravity, so that the moment about the roll axis will be such that roll when the vehicle is cornering is a minimum.

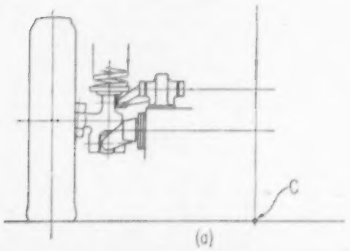


Fig. 2a. Trailing arm suspension with the pivot axes horizontal

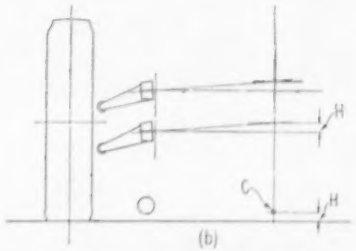


Fig. 2b. Roll centre raised by inclining the axes of the pivots

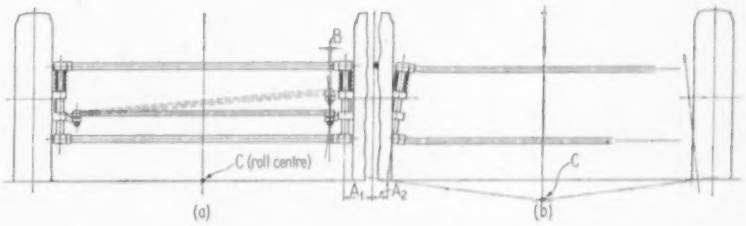


Fig. 1a. Vertical pillar suspension. Track variation, B, can be reduced by increasing the length of the track rod levers

Fig. 1b. Effect of inclining the pillars. The wheel offset, A_1, is greater than A_2 but the roll centre is low

On the other hand, it is undesirable for the roll axis to be close to, or coincident with, the centre of gravity, since under these conditions the roll may not be sufficient to give warning of impending breakaway. Some pre-war sports cars exhibited this characteristic.

The choice and positioning of road springs also has an important bearing on the subject. Widely spaced springs can provide soft suspension with good roll resistance. For this reason the coil spring is favoured. Also, it can be accommodated more simply than, say, a torsion bar and has a much better energy/weight factor than a leaf spring. The performance of a coil spring is more readily predicted and less variable than that of a leaf spring.

Vertical pillar suspension

Although it has not yet been found possible to achieve the ideal front suspension geometry, the various layouts used have their own advantages and disadvantages. The vertical pillar form of suspension, Fig. 1a, as used on Lancia and Morgan cars does, of course, give purely vertical motion of the road wheels, but unless the track rod levers are attached to splined pillars on which the stub axles can slide, any deflection from the static position of one wheel results in the road wheels toeing outwards.

If the rather expensive and complicated splined pillar arrangement is not used, the deflection must be restricted. This restriction necessitates the adoption of a higher suspension frequency than is generally considered acceptable by present-day standards. If the pillars are vertical, the instantaneous static roll centre of this system is regarded as being at ground level. If, however, they are inclined outwards at the bottom to reduce the effective offset of the wheels from the swivel pin axis, the roll centre, C, is below ground level, Fig. 1b.

Coil springs are used with this system and are mounted concentrically with the pillars. The roll resistance of this system is fairly good. This is because of the wide effective spring base, which is a feature of independent suspension systems generally, the high spring rate usually adopted in this particular layout and, with some arrangements, because of friction between pillar and stub axle member due to side loading when cornering.

Trailing arm type suspension

The trailing arm system, as used on the Porsche cars, Fig. 2a, gives a vertical motion of the road wheels as viewed from the front, but the path followed by the wheels as viewed from the side depends on the effective radius of the trailing arms. Short arms are desirable in view of the need for lateral stiffness adequate to ensure good directional stability, but, when coil springs are used, they tend to restrict the deflection obtainable. The use of longer arms helps to give satisfactory deflection, although it involves the adoption of correspondingly larger, and consequently heavier, sections to maintain rigidity; this increases the unsprung weight.

Torsion bars are a convenient form of springing, because they can be used in conjunction with short arms without there being any adverse effect on the suspension rate as maximum deflection is approached. Even this arrangement, however, has its limitations because the length available for the torsion bars is

usually restricted. This means that unless the vehicle is of reasonably low weight, the torsion bars must be highly stressed. The use of concentric or laminated torsional members has been adopted in some instances to overcome this difficulty. However, except possibly with certain laminated bars, the tendency of this is to increase prime cost.

When the axes of the trailing arms are horizontal, the roll centre is assumed to be at ground level. It is possible, however, by inclining the axes upwards towards the centre of the vehicle, to raise the roll centre by an amount equal to the vertical distance between a line joining the pivot centres on each side and the point at which the axes intersect above the longitudinal centreline of the vehicle, Fig. 2b. When torsion bars are used, it is desirable, in any case, to incline the axes in this way for two reasons. First, to reduce the angle at which the bars must enter the arms to pass each other, with working clearance, above the longitudinal centreline of the car; and also because the anchorages must be clear of the pivots. With this arrangement, some form of universal joint is usually employed between the torsion bars and the arms to prevent additional loading of the torsion bar due to bending. This is important when space and other considerations restrict the dimensions of the torsion bar.

When the roll centre is raised by inclining the axes of the links, the wheels no longer follow a vertical path as viewed from the front; so to avoid interference with the steering, the centre of the joint in the operating arm of the steering rod system must coincide with the raised roll centre.

The shock absorber arms are often employed as the upper links of this type of system. When this arrangement is used, however, the lower arms and the vertical members carrying the stub axles are usually of such proportions that they can resist twisting and bending loads without the assistance of the shock absorber arms. This, of course, involves the use of a suitable bearing between the vertical members and the lower arms. In some cases, however, the shock absorber arms are reinforced or duplicated. Despite the low centre of gravity usually obtained on cars fitted with this type of suspension, an anti-roll bar is usually found to be necessary because of the low roll centre.

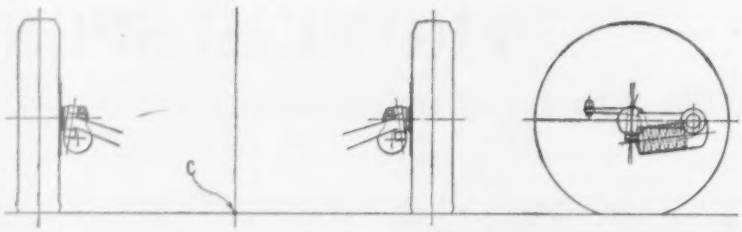


Fig. 3. The Dubonnet suspension layout, as used until recently by the General Motors Corporation

Dubonnet or leading arm suspension

Until recently, this system, Fig. 3, was used extensively by General Motors Corporation. It has similar characteristics to the trailing arm suspension in that, if the axes of the arms are horizontal, the roll centre is at ground level, and the wheels have a purely vertical motion as viewed from the front. In addition, as with the trailing arm system, the axes can be inclined to raise the roll centre, but this gives a twisting motion to the wheels as they rise and fall, although, because of the arrangement of the linkage, there is no interference with the steering. The suspension unit —that is, the spring housing or mounting, the spring, and the arm with its stub axle attached—is secured to the chassis by the kingpin. Therefore the whole of the steering linkage is sprung weight, and the advantages of this system are a very low unsprung weight and simplicity of steering linkage. One of its chief disadvantages is lack of lateral stiffness. This is particularly noticeable when cornering and is due to the space limitations of the system. Furthermore, when the brakes are applied hard, the nose of the vehicle is lifted, and this tends to give rise to unusually severe steering instability if the braking effort is not equal on both wheels. An anti-roll bar cannot be used with this arrangement.

Some of the larger vehicles fitted with this system had an additional link incorporated. This made it, in effect, a leading arm parallelogram arrangement. Such an arrangement gives improved lateral stiffness but does not affect the suspension characteristics.

Single transverse link, or divided axle, suspension

The single transverse link suspension was dealt with fully in the January 1954 issue of *Automobile Engineer*. Briefly, this form of suspension can give a roll centre as high as desired, and there is less track variation and wheel tilt than with any other system, provided the roll centre height is 70 per cent or more of the tyre rolling radius.

The greater the height of the roll centre, up to 100 per cent tyre rolling radius, the less the wheel tilt. However, wheel tilt cannot be completely eliminated, a fact that is regarded by many as a major objection to this type of suspension. Nevertheless, the author has satisfied himself that good results can be obtained, provided the R : L ratio (tyre rolling radius : distance between link pivot and tyre contact centre) is high and the wheel and tyre assemblies are of low weight. The rate of tyre wear experienced is noticeably low.

Double transverse link, or wishbone suspension

The double transverse link suspension arrangement, which is, of course, the most widely used, has many variations. An arrangement that is generally favoured is that in which unequal length links are used, with the longest in the lower position; coil springs are most frequently employed. The shock absorber arm is sometimes used as the upper link. In general, there are three ways of arranging these links, Figs. 4, 5a and 5b.

The roll centre of the arrangement shown in Fig. 4 is usually either at ground level or up to 1 in above. There is, however, a tendency for the roll centre to become lower as roll increases; thus the roll resistance is reduced when it is most needed. An anti-roll bar is therefore desirable. Track variation can be virtually zero with this arrangement and wheel tilt is also small.

When torsion bars are employed, it is usual for them to pass through, and to be concentric with, the axis of the inboard bearing of the lower link. It follows that the pivot axes of the lower links must be parallel, or nearly so, to the longitudinal centreline of the vehicle and be spaced far enough apart to provide clearance for the power unit. Torsion bars are easier to produce accurately to give the correct rate, than are coil springs; they can also be provided with a simple means of adjustment for the static position of the linkage.

Two more double link arrangements that are being increasingly used are

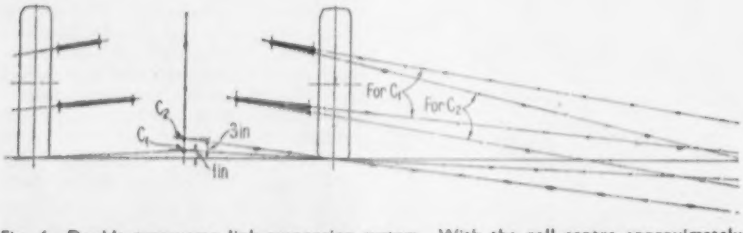


Fig. 4. Double transverse link suspension system. With the roll centre approximately 1 in above the ground level, the wheel-tilt and track variation are negligible. The links can be arranged, as shown by the dotted lines, to raise the roll centre to 3 in above the ground level. This gives a track variation of about $\frac{1}{2}$ in for 3 in bump stroke and $\frac{1}{4}$ in for $2\frac{1}{2}$ in rebound. The total wheel-tilt is about $2\frac{1}{2}$ deg

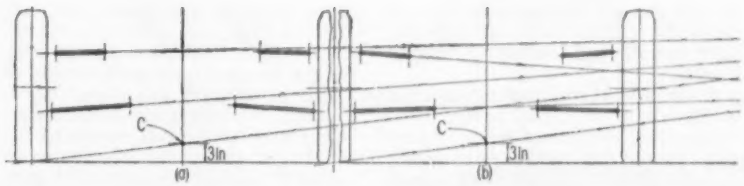


Fig. 5a. Another double transverse link layout to give a roll centre 3 in above ground level. With this arrangement the wheel-tilt tends to be less than that of Fig. 4

Fig. 5b. With this double transverse link arrangement, the track variation is approximately the same as that in Fig. 5a, but the wheel-tilt tends to be greater

shown in Figs. 5a and 5b. It can be seen that the planes of the links converge upon each other across the vehicle instead of outside it, as with the arrangement shown in Fig. 4. In none of these three layouts, however, is it possible to raise the roll centre much over 3 in from the ground without incurring excessive track variation and wheel tilt. The position of the pivots, and hence the angles assumed by the links, are critical for any given arrangement.

In general, since wheel tilt in some measure is unavoidable, the linkage should be arranged so that the wheel tilts inwards at the top as it rises to the bump position. This gives a smaller negative angle of attack (camber) of the tyres to the road surface when cornering in such a manner that body-roll is induced. The improved tyre coefficient thus obtained tends to minimize understeer and gives a better feel to the steering.

A roll centre height of 3 in above the ground is not sufficient, on most saloon cars, to permit the anti-roll bar to be dispensed with. Although in this matter the smaller cars have the advantage in that their ratio of centre of gravity height:track is generally lower than is the case with the larger vehicles.

Sweepback of double link systems has the advantage that it reduces track variation by an amount proportional to the cosine of the sweepback angle. It also makes possible the incorporation of a substantial cross member between the suspension inboard bearings, while at the same time the power unit can be installed well forward. Thus, the desirable weight distribution can be more readily obtained, and the frame has adequate stiffness at the most critical point. The importance of frame and suspension member stiffness cannot be over-emphasized, nor can the desirability of low weight.

The Ford vertical leg system

The Ford vertical leg arrangement, Fig. 6, comes close to the ideal in respect of roll centre height, and although this is obtained at the expense of some track variation on the rebound travel, the variation on the bump travel is moderate. The variation on rebound is due to the angle at which the lateral links are positioned relative to the horizontal plane, as viewed from the front. From the plan view of this suspension, it can be seen that the lower link is swept back 25 deg. This reduces track variation and provides adequately for the reaction of the drag loads.

The general layout of the system is such that the attachment points are far more widely spread than with a double transverse link arrangement, and loading on these points is relatively light. Throughout the normal range of travel, camber angle change is very small, particularly for such a high roll centre.

This system is ideal for coil spring application; the springs are installed coaxially with the legs which contain the hydraulic shock absorbers. The member that locates the wheels longitudinally also serves as an anti-roll bar. Since this system embodies a high roll centre with a wide spring base, as well as an anti-roll bar, its anti-roll characteristic is very good. It is used to its best advantage on chassisless or integrally constructed vehicles.

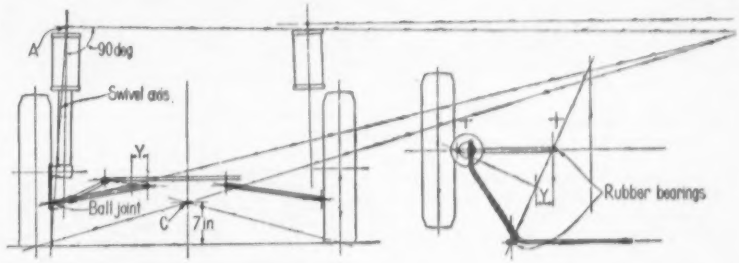


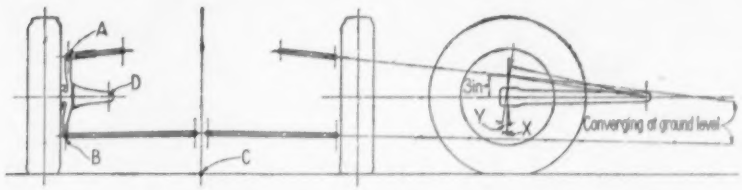
Fig. 6. Ford vertical leg suspension system. With a roll centre height of 7 in, the track variation is $\frac{3}{8}$ in for a 3 in bump stroke; for $2\frac{1}{2}$ in rebound the variation is $1\frac{1}{4}$ in. The total wheel-tilt is approximately $2\frac{1}{2}$ deg. The thrust bearing at A is in a rubber housing

The Girling suspension system

The suspension linkage arrangement, known as the Girling system, Fig. 7, is similar in some respects, so far as wheel tilt and track variation are concerned, to the double transverse link arrangement shown in Fig. 4. Each kingpin, however, is attached to a radius arm which extends rearwards and is pivoted on the frame at a point adjacent to the rear of the engine. The main advantage of this system, apart from general rigidity, is that springs of a low rate can be employed without there being any danger that the nose of the car will dip when the vehicle is braked hard. The radius arms are of such a length that brake torque transmitted through them counterbalances the load transference, thereby keeping the vehicle on an even keel.

The height of the roll centre is controlled by the lateral links and is generally approximately 1 in above the ground, as in Fig. 4. There appears to be no reason, however, why the linkage disposition shown in Fig. 5a or 5b should not be employed to raise the roll centre. In the Girling system, the castor angle is increased as the wheels are deflected. This is because the kingpin is attached to the radius arm. However, it appears to have no detrimental effect. The relatively great length of the radius arm, of course, gives a small change of angle for a given deflection, but unsprung weight is higher than with simpler arrangements.

Another aspect of the variation of the kingpin angle is that, when the vehicle is cornering, the outer wheel develops an increased castor angle due to roll. This results in increased aligning torque which would have the effect of decreasing the slip angle and gives, in consequence, an oversteer effect, which is modified to some extent by the negative angle of attack of the tyre. Also, the aligning torque of the other wheel is decreased, but this effect is



A, Ball joint (if the link can deflect torsionally, a simple pivot may be used). B, Ball joint. C, Roll centre. D, Radius arm pivot on frame. X, Castor angle in static position. Y, Castor angle increase for 3 in bump stroke

Fig. 7. The Girling suspension system

reduced slightly by the weight transference when cornering.

Conclusion

The roll centres given in the accompanying diagrams are the instantaneous static positions. These are considerably modified under dynamic conditions. The diagrams are each drawn to the same scale, the track and tyre size being common to all. In all cases an increase in track will improve the anti-roll characteristics for a given tyre size, link length, and centre of gravity height.

It should be remembered that the height of the roll centre is not necessarily a guide to the anti-roll properties of a system. For instance, if a vehicle having a roll centre at ground level is normally equipped with tyres of 11¼ in rolling radius and then subsequently wheel and tyre assemblies with, say, a 13 in rolling radius are fitted, the roll centre will be 1½ in above ground level, but the distance between the centre of gravity and the roll axis is unchanged. This matter of proportion should always be borne in mind when comparing the handling characteristics of vehicles of apparently similar suspension arrangements, otherwise misleading conclusions are liable to be drawn.

RECENT RUSSIAN PUBLICATIONS

Atlas of Soviet Automobile Design (Atlas Konstrucktzii Sovetskih Avtomobilei). Part V: Chassis

By E. A. Chudakov and G. E. Malahovsky.

MOSCOW: MASHGIS. 1954. 10½ × 20½. 204 pp. (148 Sheets of Drawings). Price 51 Roub. 80 kop. (42s. 0d.).

This is a book for engineers and technicians engaged on the design and operation of road vehicles. It provides a collection of very well produced general arrangement drawings as well as drawings of the arrangements of the main assemblies and detail drawings of a number of components of the latest Soviet six-wheel heavy lorries and four-wheel buses. The vehicles concerned are the 6 × 6 type S.I.S.—151, 4·5 ton lorry, the 6 × 4 type g.A.S.—210, 12 ton lorry, its tractor version (g.A.S.—210G) as well as semi-trailer tractor version (g.A.S.—210D), the 50 passenger (28 seats) petrol-engined inter-city bus type S.g.S.—154 and the 60 passenger (34 seated) diesel-electric city bus S.g.S.—155. Tabulated data of these vehicles give details of overall sizes, weights, turning circle, as well as engines, electrical equipment, suspension, transmissions, steering, brakes, tyres and road performance.

There are eight pages of material and heat treatment specifications for the major components of the S.I.S.—151. The material data is generally quoted in terms of the relevant Soviet specification but for some components such as frames, the steel composition is stated in detail. The design of the vehicle is shown in 57 drawings, comprising plan and elevation of the vehicle, cross sectional elevation of the chassis, detailed plan of chassis, sectional drawings of clutch and details of its main components, arrangement drawings of gearbox, details of shaft and spline tolerances and tables of gear details, details of operating gear, arrangement of distributor gearbox and splines, etc., details, arrangement of gearbox to axle drives, cardan shafts, gear change control arrangements, transmission mounting, hand (transmission) brake arrangement and details, back axle drive and details, wheels and brake assembly, rear suspension, wheel drive joints, front axle drive, front suspension, vibration dampers, front and rear brake assemblies, air brake arrangement and brake cylinders, steering gear, chassis frame, winch, and winch drive.

The material data for the type g.A.S.—210 vehicles also occupy eight pages, whilst the design details are spread over 61 drawings. In the case of the type S.g.S.—154 bus the material data occupies four pages whilst the 19 drawings are devoted to the vehicle elevation and plan, detailed section of elevation and equipment layout in plan, engine mounting, reversing switch control, cardan shaft, transmission hand brake and operating gear, axle drive brakes, suspension, shock absorbers, door operating engine and control valve. Because of an appreciable measure of standardization in the production of major components, only one page of material data and 10 drawings are devoted to the type S.g.S.—155 bus; the details concerned are the transmission control, front axle, suspension and steering gear.

The design of all vehicles is strongly influenced by U.S. practice and most of the main components have been copied direct. None of the lorries has a fully closed body, a point adversely criticized by Velikanov. The buses are of chassisless construction, both vehicles being 8 ft wide, the wheelbase and overall length being 18 ft and 31 ft, and 13 ft 5 in and 27 ft for S.g.S.—154 and 155, respectively. The engine and the 50 kW compound generator of S.g.S.—154 are mounted transversely at the rear, the 43 kW series driving motor being mounted in front of the rear axle whilst on the S.I.S.—155 the engine is forward mounted.

Generally the designs do not represent anything novel. This is not surprising since they are mainly copies adapted for Russian production methods and servicing procedure. Since there is no urge to sell, and therefore no competition, the designs, if proved suitable, are left unchanged for a long time, in order not to disturb production, spare parts supply and maintenance procedure. Thus the number of highly qualified designers engaged in the automobile industry is probably small, the main onus being on production.

The Theory of the Automobile (Teoria Avtomobila).

By G. V. Simelov, Prof., Fellow of the Academy of Artillery Sciences.

MOSCOW: DEFENCE PUB. HOUSE OF THE DEFENCE MINISTRY OF THE U.S.S.R. 1954. 5½ × 8¼. 400 pp. 247 illustrations. Price 12 Roub. 60 kop. (7s. 6d.).

This book is intended to facilitate an understanding of the dependencies and laws encountered when dealing with the motion of automobiles and to acquaint the reader with the basic methods of their determinations. In the first chapter the author deals briefly with the basic parameters of modern Soviet vehicles, that is, their overall sizes, weights, specific ground pressures, ground clearances, etc.; in addition, performance requirements are considered in general terms.

The second chapter is devoted to the rolling wheel, its mechanics in terms of rolling radius, tractive resistance and the dependence of both upon tyre pressure and load. Numerous graphs illustrate the relevant points. The next chapter deals with the equation of motion, the allowances to be made for the effect of rotary masses and the efficiencies of transmission components. Here the comprehensive data concerning the efficiency of cardan shafts as ascertained at the Scientific Auto Engine Institute (NAMI) and gearboxes under different operating conditions are of particular interest and add substantially to the knowledge of this important, though sometimes neglected, aspect of vehicle component performance. The fourth chapter is devoted to the problem of weight transfer and the coefficient of adhesion as affected by wheel load, tyre pressure, distance between dual types and road conditions.

Next, the author considers in some detail the tractive resistance. Both theoretical and practical aspects of rolling resistance as well as the various aspects of aerodynamic resistances are treated in detail and considerable use is made of German and U.S. experimental results, but without any reference to original American sources. The results of wind tunnel tests carried out at NAMI on models in the course of development of Soviet cars are of interest in supplementing already existing information on this subject.

Some 100 pages are devoted to the determination of vehicle performance characteristics, that is, the graphic and analytical determination of acceleration, climbing ability, power circulation between wheels with all-wheel drives, and retardation. All this is considered in great detail, and the clear and lucid treatment is augmented by the inclusion of worked out examples. The seventh chapter deals with the "economic characteristics," here mainly the matching of engine output and fuel consumption curves with the vehicle gear ratios, loads and general performance requirements. This important subject, which seldom receives the attention due to it, is dealt with in considerable detail.

Next, there is a chapter dealing with the determinations of correct gear ratios and the analysis of the traction and economic characteristics of infinitely variable transmissions, mainly on the basis of graphical analysis. The ninth chapter deals with vehicle stability, chiefly on a somewhat static basis along the lines adopted in classic mechanics. It is disappointing to find that no mention is made of those more recent researches in the field of directional stability dealing with the effect of tyre slip angles, the finer points of suspension design, aerodynamic characteristics and drive location.

Altogether, this is an interesting, useful and stimulating book which will meet the requirements of students and practising engineers. Its value could be enhanced by enlarging on the subject matter of the last two chapters with due regard for the development this side of the Curtain, although it might be too much to expect that acknowledgment would be made to original sources.

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

A TOOL with very great potentialities for drilling small holes is illustrated in Fig. 1. It is the Dumore automatic drill head developed by The Dumore Company, Racine, Wisconsin, U.S.A. The tool is powered by a Dumore motor, and has a Jacobs chuck mounted on one end of the armature shaft and a rotary compressor on the other. This motor performs a dual role. It supplies its own drilling power and at the same time generates its own air supply to feed the drill into the workpiece.

To operate this tool, it is merely necessary to connect the plug to any convenient supply point. Then, when the switch lever is actuated, air from the atmosphere is drawn through a feed regulator and air intake filter by the action of a rotary van compressor mounted on the armature shaft. The compressed air is then forced through a duct up to the compression chamber. This has the effect of driving the motor assembly, which is arranged to act as a piston, towards the workpiece. When the pre-set drilling depth is reached, an exhaust port opens to release pressure from the compression chamber, and a coil spring returns the drill to the starting position.

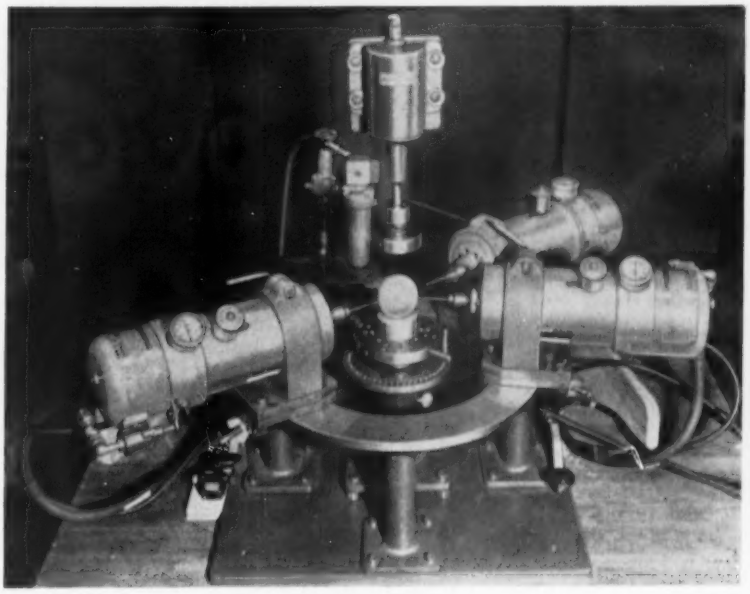


Fig. 1. Dumore automatic drill heads
(Gaston E. Marbaix Ltd.)

Depth to be drilled is pre-set by means of an adjustable stop nut over a range from $\frac{1}{32}$ in to $1\frac{1}{8}$ in. The depth can be controlled to within 0·004 in. An adjustable pressure regulator is incorporated in the unit. It allows the pressure to be adjusted up to a maximum of 15 lb/in², so that the correct pressure can be obtained for every drill size within the machine's capacity. The important fact is that the unit automatically adjusts itself to the hardness of the material. The recommended drilling sizes are: $\frac{1}{32}$ in in steel, $\frac{1}{8}$ in in brass, aluminium, zinc and cast iron, and $\frac{1}{32}$ in in plastics and wood. Dumore automatic drill heads may be used singly, in pairs, or in multiple groups. Almost any number can be wired to a magnetic contactor or relay for control from one master actuating switch. Contactors are available with from two to nine poles. The units can be so wired that all the heads operate simultaneously or in various changeable combinations.

When small holes are being drilled at depths greater than three to five drill diameters, reciprocating drilling is necessary if production is to be maintained and drill breakages avoided. For such applications there is a 20-300 repeat cycle timing device with a solenoid. Equipped with this device, the drill head reciprocates at set intervals; that is, the drill advances for a predetermined time, is then withdrawn, pauses momentarily at the top of the stroke and then advances again. This continues until the hole is drilled to depth. Up to eight units can be controlled from one repeat cycle timer. Gaston E. Marbaix Ltd., Devonshire House, Vicarage Crescent, London, S.W.11, are the agents for Dumore automatic drill heads.

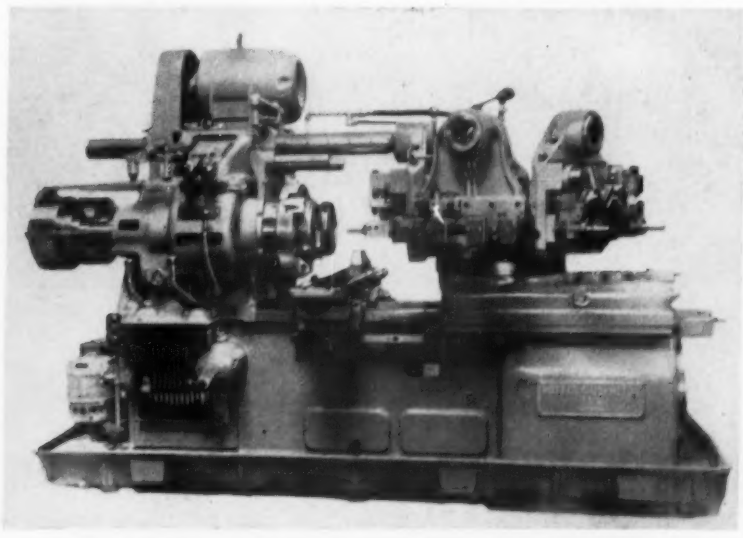


Fig. 2. Potter and Johnston two-spindle automatic chucking machine
(Burton, Griffiths and Co. Ltd.)

Automatic chucking machine

A new two-spindle automatic chucking machine developed by Potter and Johnston, Pawtucket, Rhode Island, U.S.A., is illustrated in Figs. 2 and 3. It is designated the Power-Flex type 5D 2-15 and has a distance of $15\frac{1}{2}$ in between the axes of the headstock spindles and will swing a maximum diameter of 28 in over the bedways and 18 in over the cross slide. The all-geared headstock is equipped for 28

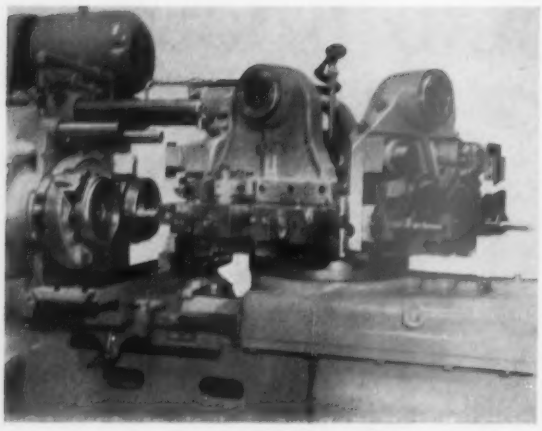


Fig. 3. The machine shown in Fig. 2 set up for first operation on rear spindle and second operation on front spindle

speed changes arranged in sets of four automatic changes. Any group of the four automatic changes may be quickly obtained by the application of the proper set of hand pick-off gears located in a compartment at the end of the headstock. The speeds range from 16 to 525 r.p.m.

There are 24 feed rates ranging either from 0·0057 in to 0·125 in or from 0·0057 in to 0·062 in per spindle revolution. These feeds are in three groups, and a feed from each group may be selected by the application of the correct pick-off gears on the end of the feed box unit. The fast motion is driven directly from the pulley and therefore is constant. All changes of speed and feed may be effected while the machine is under load. As the clutches are engaged and disengaged by power operation, the controlling dogs are not under any strain but are simply used to select the desired speed or feed. An automatic type of spindle stop, which allows the cutting tools to return to their neutral position without unnecessarily scoring the work, is provided.

The headstock spindles, which run on heavy-duty anti-friction bearings, are bored 1 $\frac{11}{16}$ in diameter and normally carry 15 in diameter air-operated chucks. Two other types of chucks can be supplied. If so desired, hydraulically operated chucks with the pump mounted on the bed at the headstock end, or electrically operated chucks with various work-holding equipment, can be fitted.

Heat-treated chrome-nickel steel is used for all the headstock gearing, which operates in a circulating bath of oil. Headstock lubricant is carried to all important points from a sump by a pressure pump. It is filtered during every passage through the headstock. The shafts for the gearings are of heat-treated alloy steel and are mounted on anti-friction bearings. As the machine has a constant speed driving pulley, the motor application is very simple, with the constant speed 15 h.p. motor mounted at the rear of the headstock and driving directly through a special multiple vee belt.

A Geneva revolving mechanism is used for indexing the five-station turret. When the indexing is completed, the turret is automatically clamped in position. The turret has two middle positions on each face for the application of tools which are in line with the axes of the two spindles. With this arrangement it is possible to perform an operation on two workpieces simultaneously, or, if the cuts involved are sufficiently in balance, it is possible to perform one operation on one workpiece and at the same time carry out supplementary work on another workpiece. Overhead pilots are standard equipment.

The turret slide is of rigid construction and has hardened and ground steel inserts to form its bearing on the base ways, which are also of hardened and ground steel. There is 16 in travel to the slide, and slide adjustment is provided whereby the distance between the spindle nose and the turret faces may be varied from 14 $\frac{11}{16}$ in to 24 $\frac{7}{16}$ in. A six-station turret can be fitted if desired.

The cross slide and base are of new design. The cross slide cam drum is located directly under the cross slide, and as the ball bearing cam roll studs are mounted in the cross slide there is a direct connection. To make it readily accessible for the application of special cams, the cam drum is carried in a compartment at the rear outside the base ways. Drive to the cam drum is through worm gearing and the entire cross slide mechanism is fully enclosed and operates in an oil bath. The cross slide unit is adjustable longitudinally for the purpose of keeping the slide in correct relation to the work held in a chuck or spindle nose fixture. This feature provides the means for keeping

Fig. 4. Deep hole drilling machine for 4-throw crankshafts
(Adcock and Shipley Ltd.)

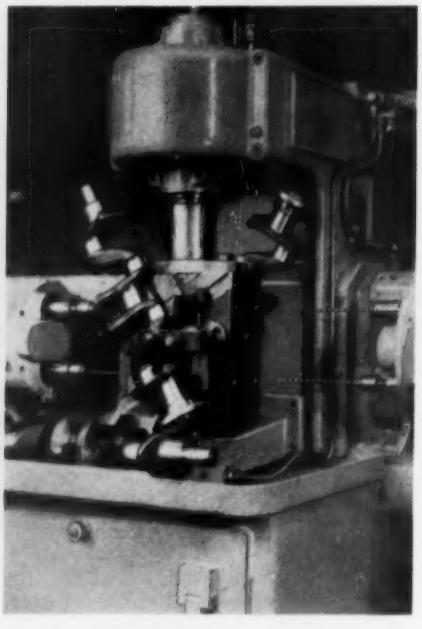


Fig. 5. Two-station indexing fixture on the machine shown in Fig. 4

the cutting pressure on the tools within the ways of the slide. If desired, the cross slide may be timed to operate during turret dwells. Burton, Griffiths and Co. Ltd., Mackadown Lane, Kitts Green, Birmingham 33, are the sole agents in this country for these machines.

Deep hole drilling machine

For suitable applications, Adcock and Shipley Ltd., Leicester, are producing hydraulic indexing fixtures for use with their deep hole drilling machines. These fixtures have been designed to allow loading to be carried out at one station while drilling is proceeding at another. This gives a considerable increase in output capacity. A machine of this type, developed for Humber Ltd. for drilling oil holes in a crankshaft for a four cylinder engine, is illustrated in Figs. 4 and 5. In Fig. 5 the two-station indexing fixture is shown with a crankshaft loaded with No. 1 throw in the upper position in the loading station, and a crankshaft with the No. 4 throw in the upper position in the drilling station.

The method of operation on this machine is to load the crankshaft with No. 1 throw uppermost for drilling holes from the main bearings to the Nos. 3 and 4 big end bearings. After this operation, the crankshaft is reloaded with No. 4 throw uppermost and the holes are drilled from the main bearings to Nos. 1 and 2 big end bearings. Therefore the component is completed in two cycles of the machine. The complete operation cycle is:—

(1) Load crankshaft, flywheel end up, at the loading station.

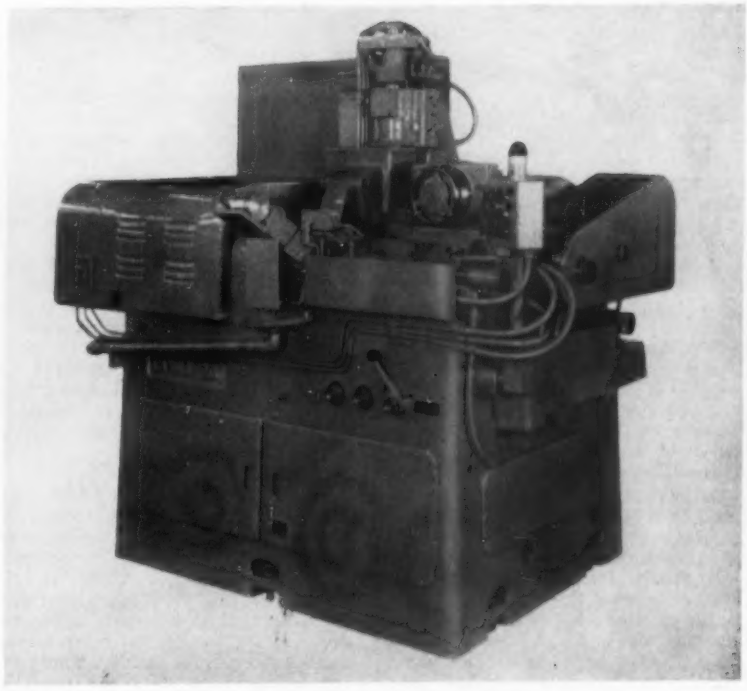


Fig. 6. Landis automatic piston grinder
(Burton, Griffiths and Co. Ltd.)

(2) Operate two handles to start the indexing. This has been specifically designed to make the operator use two hands, thereby ensuring operational safety. Immediately the fixture begins to rotate, hydraulic pressure is applied to the clamp and the crankshaft is automatically clamped in position.

(3) When the indexing is completed, the operator releases the two handles; this causes a locating plunger to engage and hold the fixture rigidly. The operator then presses the cycle start button and the drill heads go through their automatic cycle, which is:

(a) Rapid approach to work.

(b) Change to feed rate immediately before drilling starts.

(c) At a predetermined depth, rapid withdrawal to clear swarf and allow coolant to enter the part-drilled holes.

(d) Rapid approach, changing to feed rate just before the drill reaches the bottom of the part-drilled hole.

(e) Just before the drilling is completed, one head retracts while the other breaks through.

(4) While drilling is in progress, the operator loads the other fixture with a second crankshaft.

(5) When the first drilling is completed, the fixture is again indexed and the automatic cycle restarted. While the second shaft is being drilled, the partly drilled shaft is unloaded from the fixture and then reloaded with the flywheel end down.

(6) Repeat as at (5).

(7) Unload completely drilled crankshaft.

It will be seen that the total production time for these crankshafts is the time for two drilling operations plus two indexing operations. Since the holes are drilled from both ends, not only is the length of hole to be drilled by each drill shortened, with beneficial effect on drill life, but in addition the drilling time is reduced by 50 per cent.

These deep hole drilling machines are fed hydraulically by a servo-mechanism operated from cams rotated by a separate feed motor. To allow cycle times to be changed, the feed motor drive is fitted with pick-off gears. The two spindle drill heads are horizontally opposed and mounted on slides over the cabinet base. They are individually driven by 2 h.p., 1430 r.p.m. rotor stator units. The $\frac{7}{32}$ in diameter drills are fitted to standard collets, each having individual adjustment up to 1 in. By means of capstan type wheels, the drill heads have $1\frac{1}{4}$ in hand adjustment. Each head has a fast traverse rate of 300 in per minute, which is automatically changed at a predetermined point to a drill feed

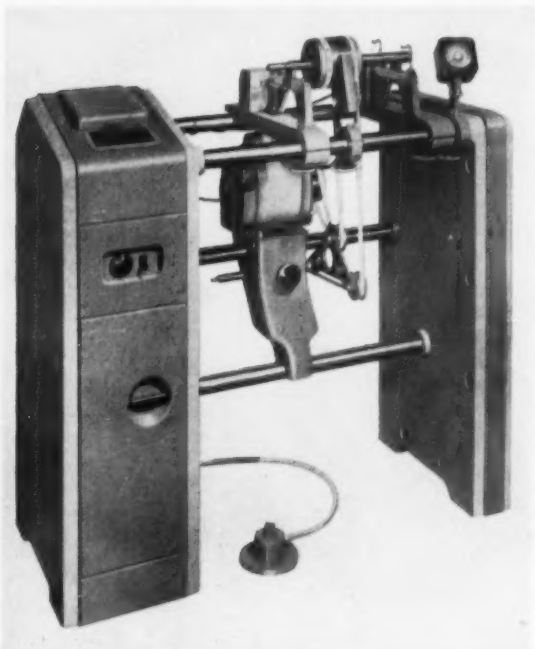


Fig. 7. Dynamic balancing machine
(Electrical and Musical Industries Ltd.)

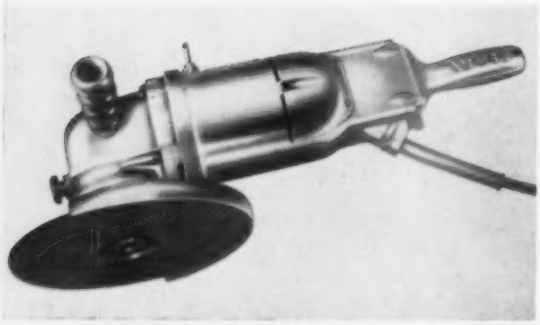


Fig. 8. AG7 portable electric heavy duty grinder
(Wolf Electric Tools Ltd.)

rate of 0·0019 in per revolution at 1020 r.p.m. Pick-off gears are provided to allow the speeds and feeds to be changed. Provision is also made for hand feeding, indexing and clamping for testing and setting up.

On the machine base, which also acts as a coolant reservoir, is mounted a box section containing the hydraulic control mechanism. Two cabinets, one at the extreme left and the other at the extreme right, are mounted on box section. The left-hand cabinet houses the electrical control gear; the right-hand one houses the hydraulic power unit. Coolant is fed from a large capacity centrifugal pump mounted in the base. It is fed to guide bushes in the fixture, an arrangement that ensures that powerful jets are directed down the holes to clear swarf and cool the work and the drills. There is a one-shot lubrication system to the drill head slides. It is operated by a foot pump near floor level at the front of the machine.

Special attention has been given to the problem of swarf accumulation. The front of the machine is equipped with a large sheet metal cabinet that allows the coolant and swarf to be collected in a specially designed swarf barrow that is housed inside the cabinet. The bottom of the barrow is fitted with grids to allow the coolant to drain out.

Automatic piston grinder

An automatic piston grinder recently developed by the Landis Tool Company of the U.S.A. for producing automobile pistons at high production rates, is illustrated in Fig. 6. It is intended that the aluminium alloy pistons should be delivered to the machine by conveyor and transfer mechanism. An automatic timed loader then places the piston in the grinding position. Work rotation starts as the grinding wheel advances rapidly to the grinding position. When this position is reached, a slow grinding feed is engaged and continues until a positive predetermined size is reached. After a timed spark-out period, the grinding wheel is rapidly retracted to the back position and the ground piston is ejected on to another conveyor. Simultaneously, another piston is brought to the grinding position and the cycle is repeated.

The pistons are ground by the in-feed method, with a wide wheel which reciprocates laterally during the grinding cycle. A master cam and rocking work cradle are used to produce the elliptical and tapered piston barrel. After a predetermined number of pistons has been ground, the wheel is dressed automatically. The wheelhead is automatically advanced the exact amount that the size of the wheel is decreased by dressing. This eliminates the need for manual compensation to return the wheelhead to the correct position for exact size.

Dynamic balancing machines

Three new models of stroboscopic balancing machines, of the type shown in Fig. 7, have been developed by Electrical and Musical Industries Ltd., Hayes, Middlesex. They are designated CS1, CS2 and CS3. The CS1 is for work from 4 oz to 5 lb weight and up to 15 in diameter; the CS2 for work from 1 lb to 30 lb and up to 15 in diameter; and the CS3 for work from 10 lb to 150 lb weight and up to 24 in diameter. These machines are designed to measure out-of-balance forces producing displacements as small as 0·000025 in. The amount of unbalance is indicated on a meter and the position of unbalance is indicated by a stroboscopic lamp.

The work to be tested is mounted in the two cradles of the machine and belt-driven from a built-in electric motor.

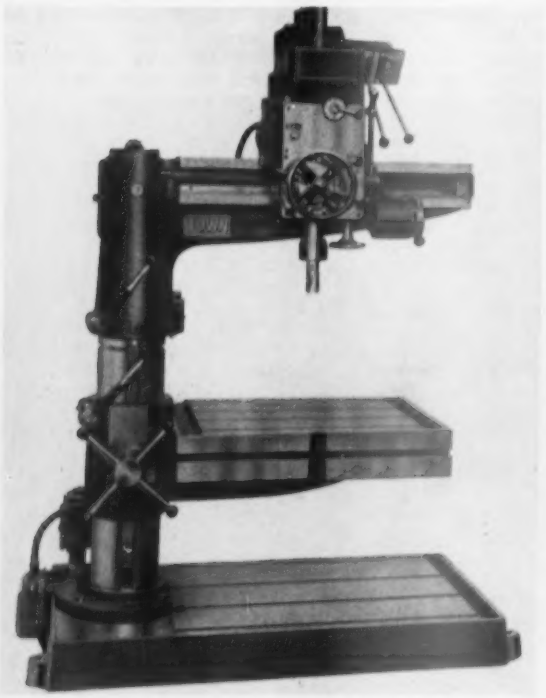


Fig. 9. Sensitive radial drilling machine
(F. Town and Sons Ltd.)

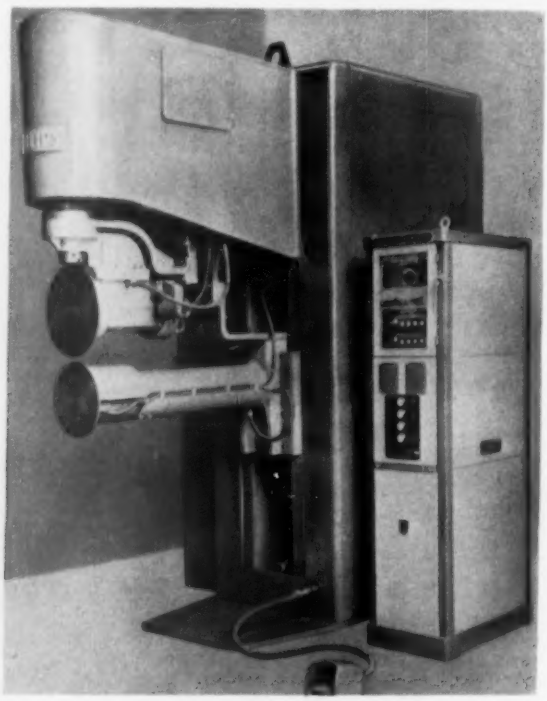


Fig. 10. Universal seam welding machine
(Philips Electrical Ltd.)

The cradles, which are free to move in one plane, are fitted with electro-mechanical transducers connected to the electronic system. This feeds the out-of-balance meter and also triggers the stroboscopic light which indicates the out-of-balance position. The transducers are completely sealed against dust and humidity, and both they and the cradle assembly can be easily changed in a matter of minutes. For high-speed production testing, automatic braking equipment can be incorporated.

Portable angle grinders

Two universal portable grinders have been expressly designed by Wolf Electric Tools Ltd., Hanger Lane, London, W.5, to operate depressed centre reinforced grinding discs. Hitherto, these discs could be used only on sanders, which do not possess the power and speed requisite for taking full advantage of the metal removal properties of the discs. The machines are based on a completely new design of electric motor, which is continuously rated at 1·5 h.p. On one machine, the AG7, which is shown in Fig. 8, the motor drives a 7 in disc at 8,000 r.p.m.; on the other, the AG9, a 9 in disc is driven at 6,000 r.p.m. These are the speeds for obtaining optimum performance and disc life.

Grinding is effected with the edge of the disc, and both machines will accommodate discs from $\frac{1}{16}$ in to $\frac{3}{8}$ in thick. The discs supplied as standard equipment are $\frac{1}{4}$ in thick, but the others are available as accessories. A $\frac{1}{4}$ in disc will suit all requirements for steel, cast iron and non-ferrous metals.

Radial drilling machine

A sensitive radial drilling, tapping and studding machine developed by F. Town and Sons Ltd., Mile Cross Works, Halifax, a subsidiary of Thos. W. Ward Ltd., Albion Works, Sheffield, is illustrated in Fig. 9. The maximum radius of the drill spindle is 3 ft 7 in and the traverse of the saddle along the arm is 2 ft 9 in. This machine is suitable for drilling up to $1\frac{1}{4}$ in from the solid in steel and $1\frac{1}{2}$ in in cast iron; the tapping capacity is $\frac{3}{4}$ in in both steel and cast iron.

The forged high tensile steel spindle is carried in a hardened steel sleeve to ensure full support at maximum traverse. Matched angular contact bearings take the drilling thrusts, and the spindle and sleeve are balanced by a compensating spring device. The feed rack is cut in the sleeve and is engaged by a special steel rack pinion. Nine spindle speeds, from 60 to 1000 r.p.m., are obtained through hardened nickel-chrome steel gears mounted on short, rigid six-splined shafts. Speed changes are effected by means of two levers. A pump in the saddle supplies oil to all working parts.

There are three rates of power feed operated by a lever on the front of the saddle. They give from 80 to 200 revolutions per inch. Any feed can be used with any spindle speed. Power feed is engaged by a slight push of the double handle on the front of the machine. The same handle gives quick hand feed when the power feed is not in use. Fine hand feed is operated by a small handwheel and sensitive hand feed by a lever at the right of the saddle. An automatic depth drilling stop, up to 4 in, is provided. It can be set in any part of the spindle traverse.

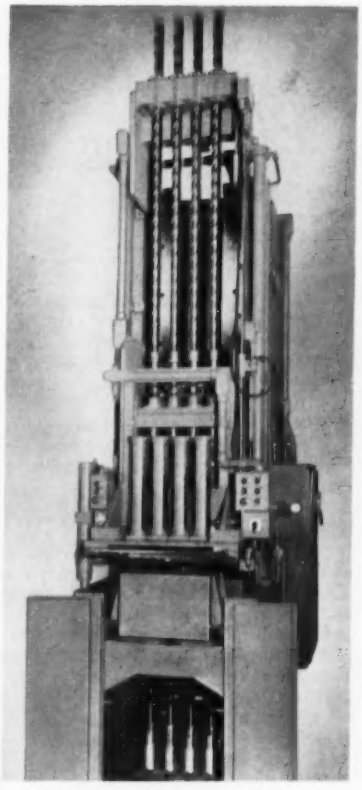


Fig. 11. Colonial broach with automatic loading and ejection
(*Gaston E. Marbaix Ltd.*)

Fig. 12. The component broached on the machine shown in Fig. 11

Seam welding machine

Universal seam welders known as the ES.2003, see Fig. 10, series, have recently been added to the range of welding equipment produced by Philips Electrical Ltd., Century House, Shaftesbury Avenue, London, W.C.2. This is a versatile machine suitable for most types of mild steel and refractory alloys. It is available with outputs ranging from 75 to 200 kVA and throat depths of 24 in, 32 in and 42 in. The welding capacity is up to 2×14 s.w.g.

A valuable feature of these machines is that they can be used for both longitudinal and circumferential welding. Change-over from one type to the other is simply effected by revolving the upper wheel assembly through 90 deg and replacing the lower assembly. Another distinctive feature is that current pick-up for the lower wheel is through a segmental commutator. This allows the wheel to run on non-conducting needle roller bearings, with a consequent increase in bearing life.

Drive to the upper wheel is through a shaft from a 3-phase motor. The speeds are infinitely reversible, forward and reverse, from 3 ft to 15 ft per minute. Vertical adjustment of the lower arm is effected by means of a hydraulic jack. The maximum height variation is 9 in. Normally, components with a minimum diameter of $7\frac{1}{2}$ in can be welded, but special arm arrangements can be supplied to suit individual requirements.

Automatic broaching

An interesting broaching application, in which two distinctly different types of spiral grooves—one an involute spline—are machined simultaneously in the same internal diameter of a transmission part is illustrated in Fig. 11. The component is shown in Fig. 12. Four stationary gear plates are broached in one stroke of a Colonial pull-up machine, and production is at the rate of 480 parts per hour. The set-up was developed by the Colonial Broach Company, Detroit 13, Michigan, U.S.A.

During the 60 in broaching stroke, the internal diameters of four gear plates are broached to 1·389 in diameter, and nine involute internal spiral splines and three internal helical grooves are cut. The four-station fixture is equipped with four vertical loading tubes and an automatic shuttling device with a $10\frac{1}{2}$ in stroke, which receives the parts from the loading tubes and locates them in position for broaching. The broaches are drawn through the work in a spiralling motion produced by helical lead bars attached to each broach puller.

After the shuttle fixture is retracted, the broached parts drop on to a hinged baffle that directs them into an unloading chute. The broaches are then retracted and the cycle is automatically repeated. A special hydraulic circuit, synchronized with the machine cycle for automatic operation, actuates the shuttle fixture. The complete machine cycle, including loading and unloading, takes 30 seconds, and with the broaching machine operating at a cutting speed of 15 ft per minute, eight parts are completed every minute. Gaston E. Marbaix, Ltd., Devonshire House, Vicarage Crescent, London, S.W.11 are the British agents for Colonial broaching machines.

TEST LABORATORIES

Methods and Equipment Employed in the Fighting Vehicle Research and Development Establishment, Ministry of Supply

H. R. Swatman, A.M.I.Mech.E., M.I.R.T.E., A.M.Inst.R.

THE ultimate method of proving a fighting vehicle or heavy and bulky equipment must be by field trial and operational use in every case. For much of the development work leading up to the design and installation of any particular equipment, as well as for *ad hoc* investigations on causes of failure or unsatisfactory operation, fully controlled rig and laboratory test conditions are always a great advantage and often essential. This is particularly so for such as the following reasons: (a) when an operational run of several hours duration may be needed to establish steady conditions; (b) when inaccessibility makes observation of the test component impossible after installation; (c) when great distances are involved in transporting to natural test locations, vehicles, equipment, supplies and test personnel; (d) waiting for suitable testing conditions, with possible early failures or short duration of tests involving waste of time; (e) inevitable lack of close co-ordination between the design establishment and manufacturers due to distances, costs involved, and time delays after required modifications and redesign have been specified.

With these needs in mind, the establishment of engineering test laboratories in the Fighting Vehicles Research and Development Establishment was embarked upon early in the war. At first it was necessary to concentrate full attention on urgent short-term testing requirements, but as the available facilities developed it was possible to undertake investigations of a comparatively long term character. Since the end of the war attention has been turned further towards the solving of basic problems, using more refined techniques.

One field in which laboratory tests have been found particularly successful is that of climatic trials in extreme conditions, which is described in more detail later in the paper. The laboratory facilities include provision for the exposure of specimens, ranging in size up to a complete armoured "Conqueror" tank, to controlled, simulated arctic and tropical conditions.

Work at the experimental wing, Farnborough

When early in 1941 it was decided to establish testing laboratories in the Department of Tank Design, little was available in the way of equipment or other facilities, and it was necessary to build laboratories, obtain and install equipment and recruit labour. The technical staff appointed comprised an assistant director, a senior engineer as his deputy (the author of this paper) and three very junior technical assistants. The only equipment available consisted of two Heenan and Froude hydraulic water brakes of a very old type, with some dilapidated miscellaneous instruments and panels transferred from a disused test shop at Woolwich Arsenal. This equipment was installed in a hangar at the D.T.D. experimental wing, Farnborough, and was to be the first F.V.R.D.E. test house.

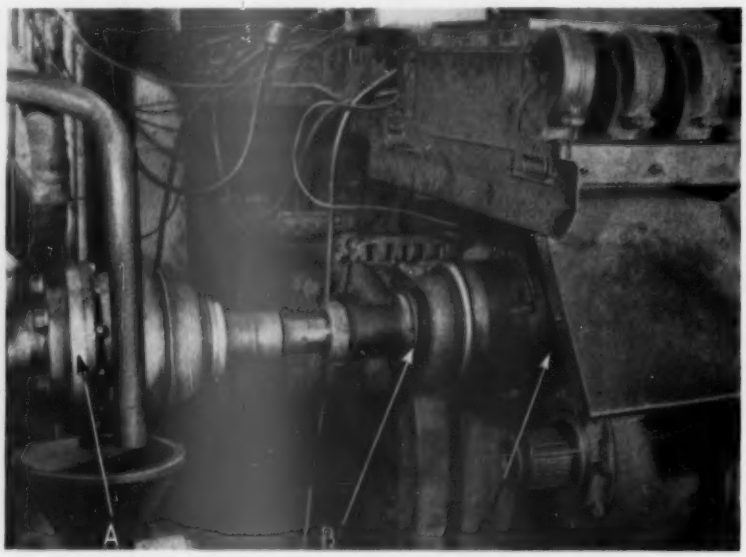
In the meantime, arrangements were made for a supply of modern equipment to be installed in a new building, then under construction at Longcross, to cover the types of test work envisaged. In addition, contracts were made with various firms and organizations, to determine to what extent their facilities could be used or loaned for tests on behalf of the department. The assistant director undertook these activities and the greater part of the organization of the engineering test laboratories, and the immediate test work, was at this stage the responsibility of the author.

The latter part of 1941 saw the commencement of actual, operations, which began as qualitative *ad hoc* tests on small components, and later developed into work on suspensions, engine cooling, fan performance and power output measurements on complete vehicles. Among the firms and organizations which assisted the Department with test work at this period were Leyland Motors Ltd. (engine air cleaners), A.E.C. Ltd. (heat distortion of brake drums) and the I.A.E. research department (fatigue tests on suspension units).

Work at Longcross site

From the inception, the arrangement had been that test rigs were designed in the Engineering Research Laboratories and constructed in the Fighting Vehicle Proving Establishment workshops at Farnborough. The facilities available in this direction were greatly extended when the new F.V.P.E. workshops at Longcross were completed. However, there were still some difficulties because the Engineering Research Test Laboratory buildings and much of the equipment were still unfinished. As this position eventually smoothed out it became possible to embark upon a number of test and research investigations and to transfer the Farnborough test house equipment to Longcross, early in 1942.

Simultaneously with the erection of the main Engineering Research Test Laboratory, a scheme was proceeding for the provision of low temperature test facilities simulating arctic conditions and capable of handling complete large armoured vehicles or bulky equipment to serve the



A. Dynamometer coupling B. Cardan shaft C. Vehicle final drive shaft

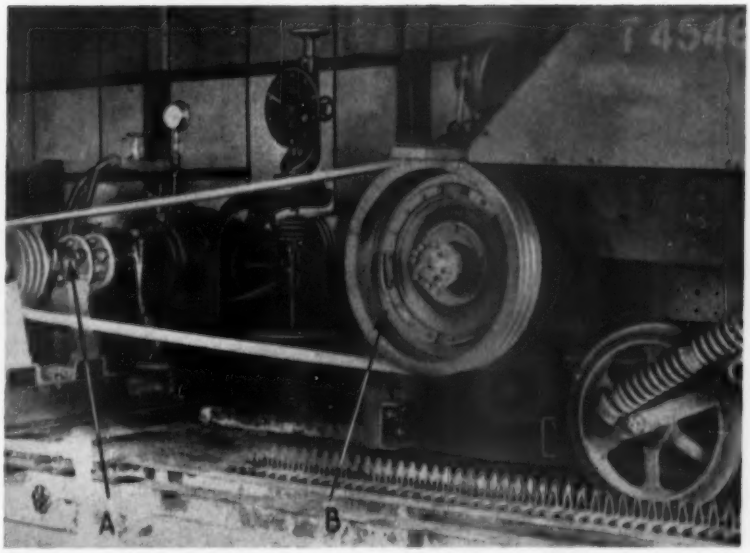
Fig. 1. Set-up for determining power output at the driving wheels

needs of all departments of the Ministry. This laboratory was completed during the early part of 1943. At this juncture during the growth of the laboratories, organization, and when urgent demands for research investigations and tests were rapidly accumulating, the assistant director in charge of the branch resigned from the Establishment and the author was appointed Superintendent of the Laboratories.

Later in the building development programme, the author prepared a scheme for the provision of test facilities to enable vehicles and equipment to be subjected to simulated tropical conditions. This tropical laboratory was completed in time to meet the test demands for design information requirements as the war theatre extended to the tropics. Subsequently these laboratories have been in continuous use for both military and commercial development investigations. These extreme climatic laboratories are described later in this paper.

During the war practically the whole attention of the Engineering Research Test Branch was necessarily devoted to dealing with tests in connection with day-to-day problems. At this time designs and ideas were changing rapidly and, as questions and problems were put before the Engineering Research Test Laboratories, the answers and solutions were urgently required and expected immediately. As design problems became stabilized and the facilities in the laboratories were extended and became available, it was possible to undertake research and development work of a comparatively long term nature. On the presumption that the future would bring further expansion to this side of the work, the designing of rigs and permanent test equipment was left to the choice of the Engineering Research Test Laboratories Branch, and was pursued with these requirements in mind.

After the end of the war the Wheeled Vehicle Design Department of the Ministry of Supply was incorporated in the Fighting Vehicle Research and Development Establishment. The wheeled vehicle test facilities, which included a tyre and wheel test development laboratory and a vehicle chassis dynamometer test house, were placed under the supervision of the Superintendent of the F.V.R.D.E. Engineering Research Test Laboratories. The laboratories have provided facilities for all the specialist branches of F.V.R.D.E. on a "common service" basis. In addition, the Superintendent of the laboratories was authorized to employ commercial firms' facilities, if necessary, by placing of extra-mural research contracts and, when the F.V.R.D.E. facilities were available, to carry out research for other Government departments and commercial industry on a cost recovery and charge basis. Reports on the work carried out are issued with comments and conclusions to the department or branch initiating the work. If the reports are given general circulation they are issued with a



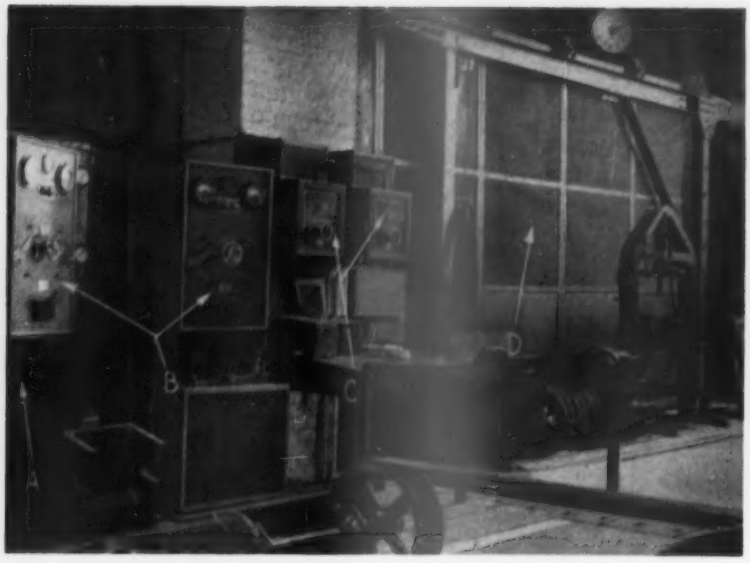
A. Dynamometer extensions B. Vee pulleys replacing driving sprockets

Fig. 2. Another set-up for determining power output at the driving wheels

A. Vehicle to dynamometer cardan shaft B. Vehicle test tunnel

Fig. 3. The original vehicle test tunnel at the F.V.R.D.E.

Fig. 4. Air inlet louvres for the re-designed and modified tunnel



A. Original tunnel B. Dynamometer control panels C. Variable speed fan control D. Telescopic extension

Fig. 5. Exterior of the new test tunnel

signature of approval by the Chief Engineer, F.V.R.D.E.

By this process of evolution the present organization was built up. Broadly, it comprises technical staff who, in consultation with the design engineer and user, devise the experimental methods, conduct the research investigations, record the test observations, analyse results and prepare reports, and workshop staff who carry out the construction and erection of rigs and prepare equipment for testing. It was proved advantageous to provide not only a small design and drawing office but a machine shop integral with the laboratories' organization, although as previously explained, the heavier test equipment was either purchased complete from the makers or constructed to the Engineering Test Laboratories, designs in the workshops of the Fighting Vehicle Proving Establishment.

As previously mentioned, the test techniques now in use are based in great part on the results of past experience. In the description of the available facilities which follows, an indication is given in some cases of the development of the technique in question and of any lessons learned in the process. The equipment is described under three general headings as below:—

(a) Permanently installed test plant for general application.

(b) Instruments of a portable nature which can be incorporated in different test layouts as required.

(c) Specialized test rigs constructed to satisfy particular test conditions.

A. Cardan shafts to dynamometers B. Extractor fans C. Mock-up of vehicle engine compartment

Fig. 6. Interior of the test tunnel shown in Fig. 5

Permanent test equipment

Vehicle dynamometer plant. One of the earliest problems for which facilities were urgently required was the running of a vehicle under steady ambient temperature conditions at full power, in order to measure the output at the driving wheels or sprockets, and to investigate the effect of modifications to cooling or silencing systems. Initially, work of this nature was carried out with one track removed from its driving sprocket (in the case of a tracked vehicle) and the sprocket or driving wheel coupled to a Heenan and Froude brake by means of cardan a shaft, see Fig. 1.

In some cases the other sprocket or driving wheel was locked and in others the differential mechanism was removed and a solid bevel drive substituted, so that in any case the whole power output was delivered on the side connected to the dynamometer. This method involved making a correction for the power loss in the differential gearing and was later modified by placing the dynamometer behind the vehicle and connecting both sprockets or driving wheels to extensions of the dynamometer shaft by using vee-belt drives, see Fig. 2. These methods provided a ready means of assessing the power output and cooling performance of experimental and production vehicles, and ultimately became a basis for vehicle proving and an essential requirement for the guidance of design branches in the development of new vehicles.

Running a vehicle stationary in an open building, however, proved far from satisfactory. In the first place, the ambient conditions were difficult to stabilize and in any case the terminal ambient temperature was too high to represent normal field conditions and air. Moreover, recirculation was a difficult and confusing variable during the test. To overcome these difficulties a vehicle test tunnel was designed, with provision for continuously drawing air over the vehicle from front to rear and expelling air from the building. This provided ambient conditions approximating to those experienced with the vehicle in motion in the open, and furthermore had the additional advantage of removing objectionable fumes and heat from the buildings.

The first arrangement used at Longcross was a short tunnel about 12 ft square in cross section and 20 ft long, provided with two propeller suction fans at the outlet and drawing air from the interior of the building, see Fig. 3. This scheme did not give completely steady test conditions, and moreover, produced undesirable air disturbances in the building. The tunnel was therefore redesigned and modified so that air was drawn direct from outside through louvres in the opposite wall

of the building, the tunnel was widened to 15 ft, was extended telescopically up to the louvres and can be folded when not in use. Vehicles up to 50 ft long and 13 ft wide can be accommodated. This tunnel is shown in Figs. 4, 5 and 6.

Separate dynamometers were provided to absorb the power from the two driving wheels or sprockets. The dynamometers, which were mounted on adjustable pedestals at right angles to axis of the tunnel, were of the Heenan and Froude E.C.L.5 eddy current type, and each was capable of absorbing up to 500 h.p. at speeds between 300 and 2,500 r.p.m. They were so arranged as to be capable of separate operation for the direct testing of power units or transmission components. As an illustration of what can be achieved by simple means with a rig in a laboratory, the b.h.p. curves shown in Fig. 7 were obtained on a wheeled chassis placed in

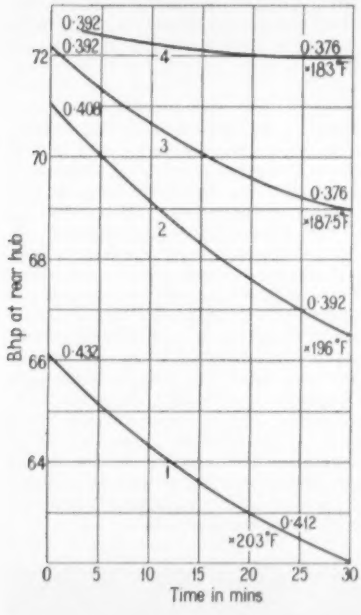


Fig. 7. B.H.P. curves for a variety of conditions. End temperatures of cooling water are shown at X

the wind-tunnel, with the rear axle jacked up, so that the power output at the rear hubs could be recorded.

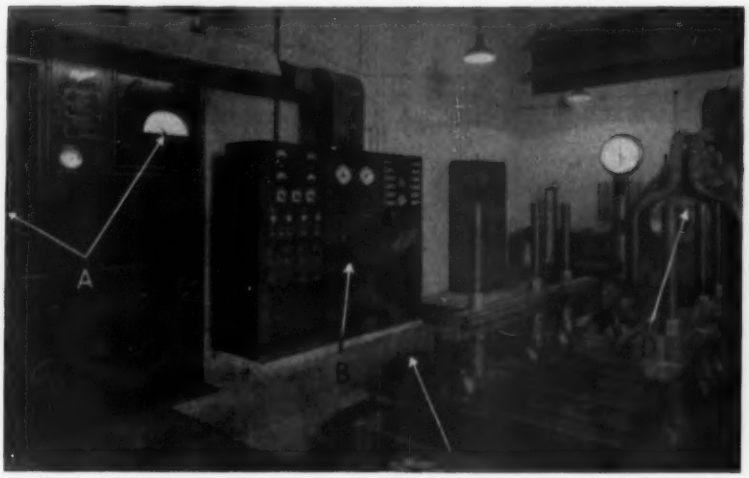
Curves 1-4 were obtained, starting in each case with a cylinder jacket temperature of approximately 120 deg F, and a gearbox and axle temperature of approximately 100 deg F. Operating conditions were:— full throttle, 2nd gear, 1,200 r.p.m.

Curve 1 is the result with S.A.E.30 oil in the engine and S.A.E.90 oil in the gearbox and axle.

Curve 2 is the result with S.A.E. 5W oil in engine and gearbox and thin synthetic oil in the axle.

Curve 3 is the same as curve 2, but with induction air at ambient instead of at under-bonnet temperature.

Curve 4 is the same as curve, 3 but with fuel pump also cooled to ambient temperature.



A. Fuel measuring and weighing apparatus B. Control and instrument panel
C. Water cooled exhaust system D. 1,000 h.p. dynamometer with 275 h.p. motor and clutch

Fig. 8. One of the test cubicles

The slope of curve 4 will obviously depend upon the mechanical condition of the fuel pump and injectors. It will be seen that, when comparing curves 1 and 4, the overall effect of using thinner lubricants and reducing inducted air and fuel pump temperatures, increases the power at the rear hubs by 10 h.p., with an engine cooling water temperature 20 deg F lower. By setting back the fuel pump delivery stop when operating under the conditions specified for curve 4, so that fuel delivery limits the power output to that shown on curve 1, the result obtained represents a very considerable gain in fuel economy under full-load conditions for the same vehicle performance. Alternatively, by leaving the pump stop at normal setting, a gear ratio could often be saved under hilly route operation. This would decrease engine revs per mile and so save fuel. The specific fuel consumptions in lb/b.h.p./hr at the rear hubs are shown above the curves.

Engine test equipment

With a view to maintaining steady test conditions and localizing noise and fumes, the two main engine test beds were located in cubicles fitted with folding doors. One cubicle was equipped with a Heenan and Froude hydraulic brake capable of absorbing up to 600 h.p. at 900 to 2,700 r.p.m., while the other was equipped with a motor and eddy current dynamometer system absorbing up to 1,000 h.p. and motoring at powers up to 275 h.p. over a speed range of 500 to 3,000 r.p.m., see Fig. 8.

Both cubicles were provided with panels centralizing all controls and permitting ready observation and recording of variables such as speed, torque (braking or motoring), pressures,

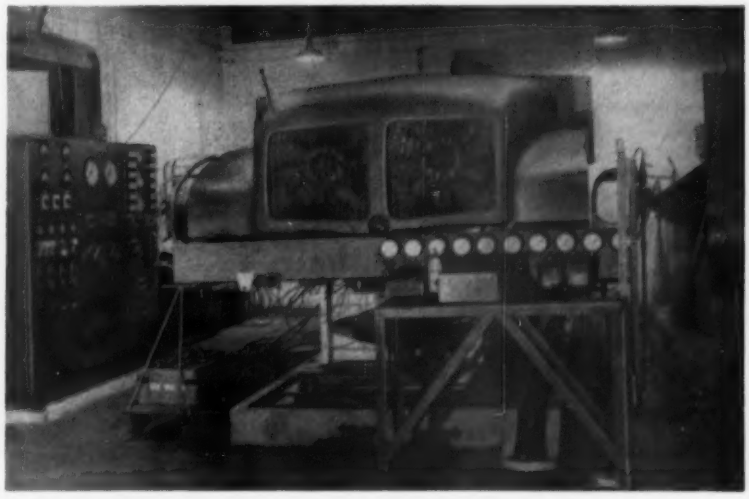


Fig. 9. Prototype engine compartment installed for test

temperatures and fuel consumption (oil or petrol). Suitable safety interlocks were also incorporated. Softened cooling water was supplied from a self-contained Heenan and Froude system with its own external cooler, precise control of water temperature being obtained by admission of cold water from the main supply. Provision was made for a cylinder cooling water system operating under pressures up to 25 lb per sq in, and water cooled exhaust pipe trenches were also included. Fan ventilation systems were installed capable of giving roughly three changes of air per minute in both cubicles.

These test beds can be used either for testing engine and transmission components separately or for the proving of complete engine compartment mock-up systems, see Fig. 9. It is a test laboratory and design practice to build a complete mock-up of the vehicle engine compartment including the transmission cooling fans, radiators, louvres, air cleaners, petrol tanks, pipes and all fittings as installed in a complete vehicle, and to run trials simulating conditions which will be obtained in a vehicle at full power or maximum torque.

In addition to the two main engine test cubicles, two further test beds for smaller engines were laid down. These were equipped with Heenan and Froude water brakes capable of absorbing, respectively, 400 h.p. at 2,700 r.p.m. and 225 h.p. at 3,000-6,000 r.p.m. Cooling and fuel services were taken from the general system serving the cubicles.

The performance of the recording equipment in the engine test cubicles has been found generally satisfactory, with the exception that the indications of the petrol consumption meters, which are of the calibrated orifice type, have been found liable to vary with ambient temperature. It was therefore found advisable to replace these by meters of the calibrated reservoir type where the flow can be timed by stop-watch. Experience has also shown that thermo-couples, used with a potentiometer, are more satisfactory than the mains-operated resistance thermometers installed in the panels. In order to obviate errors due to bimetallic contacts at the selector switch, both the hot and cold junctions must be arranged on the same side of the switch in the thermo-couple circuit.

Wheeled vehicle chassis dynamometer test

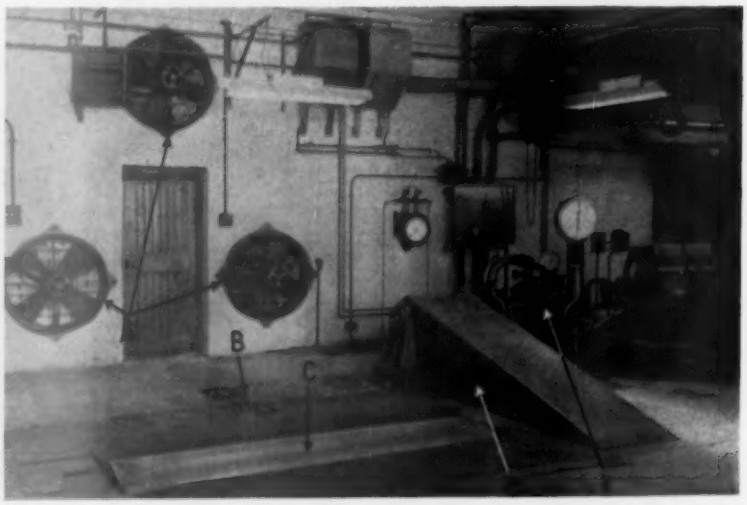
The technique of testing wheeled vehicles and chassis on a drum dynamometer rig is not revolutionary. By positioning and anchoring a vehicle at right angles to the axis of a drum so that the axle and road driving wheels are centred and in friction contact on the drum periphery, it is possible, by using the vehicle power unit, to drive the wheels and rotate the drum. Then, by incorporating a suitable drive from the drum to a dynamometer, vehicle performance and output at road wheels can be assessed. There are certain limits and disadvantages, however, in using this technique for a great amount of vehicle evaluation, due to tyre slip on the drum, belt slip or other drive frictional losses between the drum and dynamometer, and the installation conditions under which the rig is being operated. Tyre slip on the drum and the frictional or slip losses between the drum and dynamometer can be assessed but to do so involves a great amount of time and inconvenience.

The installation of the rig and test procedure can be made sufficiently satisfactory to enable a considerable amount of research and experimental work to be carried out in order to solve problems which would be extremely difficult to investigate in the field. This is particularly so where complicated instrumentation, continuous endurance runs, repeatable conditions of test and carrying out modifications without removing the chassis or vehicle off the rig, are essential. Vehicle chassis dynamometer testing enables problems to be investigated and aids development research on (a) engine, gearbox and axle performance and transmission losses, (b) radiator and cooling fan efficiencies, petrol vapour locks, oil seal investigations, and under-bonnet air recirculation and dissipation. In many cases the costs of the tests are lower than field trial investigations and the test programmes can be accelerated. The installation and rig comprise a 5 ft diameter by 8½ ft wide drum mounted in a frame and built in a pit so that the top of the drum is level with the floor of the test house. A Heenan and Froude D.P.Y.5 dynamometer was positioned on the floor, set back and parallel to the axis of the drum with a multiple "vee" belt drive from the drum to the dynamometer. At the rear, and also in front of the rig, are anchoring down clamps and channels with tie bars to secure a vehicle in position. The drum is capable of carrying a 10-ton dead weight axle load and can be rotated at periphery speeds up to 300 r.p.m., that is, approximately 50 m.p.h.

To simulate the conditions of a vehicle travelling on a road, a blower unit with a 30 in diameter fan can be positioned to blow cold or hot air at 5-18 m.p.h. on to the radiator and bonnet of a vehicle on the rig. Controlled speed and air extractor fans installed in the rear of the building draw air over and past the vehicle. The building is a brick and steel structure with steam radiators to enable temperatures simulating tropical conditions to be obtained. Banks of infra-red lamps are available to simulate solar ray effects on vehicles and equipment, see Figs. 10 and 11.

Tyre testing plant

The disadvantage of field tests, whether short or long term, when carried out on the public highway or on a test track, is that it is difficult to maintain the high road speeds and impossible to control climatic conditions necessary to attain the high tyre casing temperatures for the purpose of comparative proving and development of tyres. Where a test period exceeding normal running hours is desired, and this is often the case, the test is invariably interrupted by inclement weather and advent of night time. To overcome these difficulties and substantially decrease the test period and cost, tests are carried out on indoor machines. Thus, 24 hour or longer test periods can be run without difficulty. Conditions in indoor machine testing of tyres can be maintained constant, whereas in the case of road tests, road and test conditions are continually changing. The machines which are described below have been of real value and have expedited the solution of tyre design and development problems for military purposes. The tyre testing plant and equipment comprise:—



A. Air extractor fans B. Anchoring down clamps and tie bars C. Drum 5 ft dia × 8 ft 6 in wide
D. Cover over multiple vee belt drive E. Dynamometer

Fig. 10. The installation for vehicle chassis dynamometer tests

(a) Two machines capable of subjecting a tyre to load conditions of 3 tons, and one machine of 10-ton capacity. The 3-ton machine accepts one tyre which is driven in a vertical position on the drum drive. The 10-ton machine can test two tyres simultaneously when the tyres are positioned horizontally. Tyres up to and including the 11.00—20 size can be tested on the machines at speeds simulating 10 to 80 m.p.h. These machines are dead weight loaded and are so arranged that electric dynamic braking of the drums automatically occurs directly the loading beam position is effected owing to tyre failure or pressure decrease. This automatic brake is sufficiently quick in action to stop the machine and prevent the tyre running deflated. Running the tyre deflated would make the cause of the tyre failure more difficult to assess.

These three machines are equipped with multivane blowers complete with 7½ h.p. electric motors and starters. The blowers can be directed on to the tyre zone by means of suitable trunking which is fitted with swinging dampers to control the speed of air provided by the blowers. Provision is also made for the bleeding of tyre pressure increase whilst running under test, thus maintaining the initial pressure and obviating the need to stop the test for this purpose. Rotary joints are connected to pressure gauges to record the conditions of inflation pressure in the tyre on test. Recording ammeters are connected in the electrical circuits between the electrical motors and the driving drums to record the current required at any time to drive the tyre undergoing test.

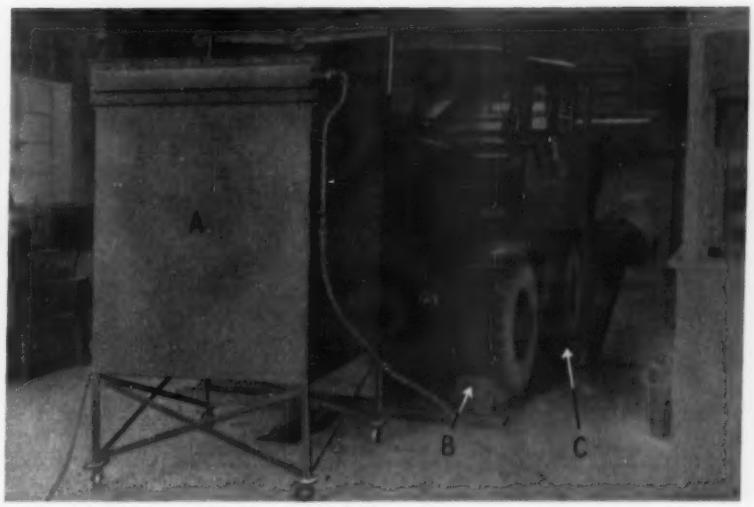
(b) A tyre impact machine for investigating damage caused to tyres by sudden blows and impact with sharp or raised objects, such as kerbs, etc.

(c) A tyre deflection beam installed in conjunction with a 10-ton weighbridge to enable tyre deflection at maximum load to be assessed.

(d) Service and general equipment, including an air compressor giving an output of 20 c.f.m., complete with a compressed air storage tank and fittings, a tyre inner tube testing tank, air temperature and pressure recorders etc., see Figs. 12, 13, 14 and 15.

Hull, chassis, suspension and wheels test rig

For controlled testing on tank hulls, vehicle chassis, suspension units and wheels under dynamic load, a comprehensive rig was devised. Essentially, the rig consists of a pair of 4 ft diameter by 3 ft wide drums driven by a 40 h.p. motor and capable of giving peripheral speeds equivalent to 5 to 75 m.p.h. The wide variation of drum speed was obtained by using in the drive to the drums a Covenanter tank gearbox, which incorporated the Wilson epicyclic steering unit and thus gave the smooth take-up necessary in view of the high inertia of the drums. The whole rig was erected in a pit so that the running surfaces of the drums were at ground level. Access for vehicles was given by detachable ramps so that, if necessary, a complete vehicle could be positioned over the rig to allow tests simulating, as near as possible, running conditions to be carried out. The wheels undergoing test were friction driven by the drums, and cams representing the inequalities of ground or road could be fitted to the drums. Loads not exceeding up to 10 tons on each drum could be applied when subjecting suspension units to test, see Fig. 16.



A. Hot or cold air blower unit. B. Wheel chocks C. Driving wheel positioned on drum

Fig. 11. The set-up to simulate the conditions for a travelling vehicle

Cooling fan test tunnel

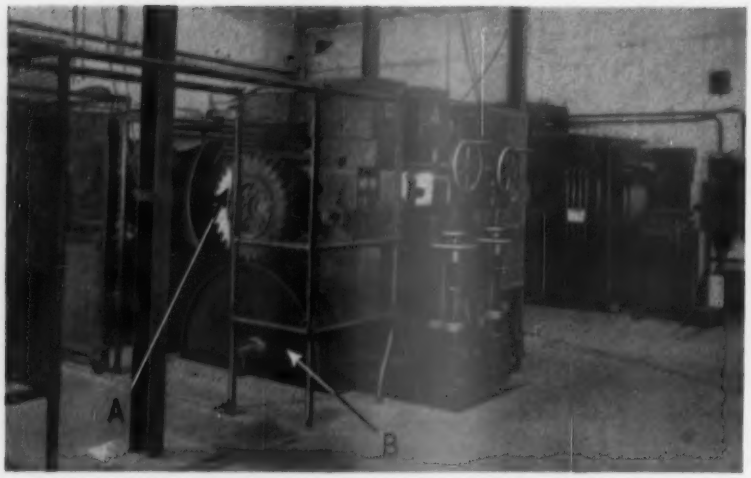
Comparison of the performance and power requirements of engine cooling and ventilation fans were made in a test tunnel designed in accordance with British Standard Specification No. 848-1939. The power input to the fan was measured when driving the fan with a motoring dynamometer using a transmission with a known power loss. Air delivery was measured by traversing a pitot tube across the cylindrical duct, at the inlet end of which the fan was mounted. After experience gained in the use of the equipment, it was proved advantageous and possible to neutralize the power loss by mounting a gearbox integral with a swinging dynamometer carcass. The method of measurement of the air delivery has been considerably improved by the use of fixed pitot combs used in conjunction with side tappings made in the wall of the tunnel. An air flow straightening device was fitted downstream between the fan and the pitot comb. The restriction against which the fan was operating was varied as required by adjusting a conical damper positioned at the outlet end of the duct, see Fig. 17. Fans having a diameter from 16 to 20 in may be tested at speeds up to 4,000 r.p.m. The fan test rig has also been used for investigations on the noise level of fans and the operating life of fan belts.

To illustrate what can be achieved with the fan testing rig, the tests on an experimental engine cooling fan to assess fan performance and to obtain total head and velocity head recordings were as follows: The fan had eight equally spaced blades of Aerofoil section, the blade angle and chord length being constant from tip to root. The blades were made adjustable about their radial axes to permit an investigation and to assess fan performance at a number of predetermined blade angles.

Tests were made to ascertain the performance of the fan with a blade tip to cowl clearance of 0·25 in and blade angles of 25, 30, 35, 40 and 45 deg. Test readings of total head and velocity head were recorded at a fan speed of 2,000 r.p.m. under various orifice conditions from fully open to fully closed. Ambient air temperatures and barometric pressure were recorded after each test run, enabling the volume of air under standard conditions to be calculated. Dynamometer loads were recorded to allow the horse power absorbed by the fan and the efficiencies to be calculated.

Engine air cleaner test cabinet

The original air cleaner test rig was supplied by British Filters Ltd. After the experience gained in its use and with the knowledge of the requirements of future air cleaner development, the Test Laboratory staff designed and built a new rig and improved the testing technique. The cleaner under test was mounted in either a level or a tilted position in the chamber and air charged with test dust drawn through it by means of a fan which exhausted to the open air. The standard dust/air ratio is 0·04 grammes per cubic foot of free air. Air enters the chamber through a pipe in which the flow through the cleaner under test is measured by means of an axial pitot tube, the pipe having been calibrated against a standard orifice plate (three pipes of different flow capacities were provided on the



A. Tyre mounted for testing B. Driving drum
Fig. 12. 3-ton tyre testing machine

rig). The restriction set up by the cleaner was measured by a manometer connected to an orifice in the wall of the pipe leading to the suction fan.

For tests on fighting vehicle air cleaners, the procedure followed was laid down in A.F.V. Specification No. 1040, which permits tests to be carried out by direct feeding of the dust or testing in an artificially produced dust storm. It also specifies the size and analysis of the test dust, which was intended to be representative of air-borne dusts met with in various parts of the world, with particle sizes ranging from what would pass through a sieve of 100 mesh down to a few microns. The dust actually used was made up by adding together suitable proportions of Cooke's Grade "C" sand and the abrasive powder known as "Abra" (a brand of air-floated Cornwall silica) and the specified tests were devised to determine the cleaning efficiency at various air flows and rates of restriction. The particle size distribution of the test dust which was ultimately incorporated in the latest B.S., S.P.20 and B.S. 1701 (1950) was formulated as a result of analysis of numerous samples of dust recovered from air cleaners taken from vehicles which had been operating under various and extremely dusty conditions.

Originally the dust feed supplied with the test rig comprised a conical hopper with a needle valve controlling the flow to a chamber from which the dust fell through a number of tubular jets on to the curved surface of a cylindrical drum rotating about a horizontal axis, and thence into the test chamber; the jets were kept in vibration so as to reduce their tendency to clog. This arrangement was unsatisfactory, especially at low rates of feed, in view of the fine adhesive and packing nature of some of the dust particles. The development of a device for feeding test dust at constant rates involved a great deal of experimentation. After a number of modifications had been made, an arrangement was eventually proved satisfactory and put into regular use. The mechanism finally adopted was a rotating plate and scraper type of dust feed by which dust is delivered from a funnel to a rotating dished plate. The funnel, which was heated with a carbon filament lamp to keep the dust dry, and incorporated a vertical tongue, was vibrated intermittently to ensure free flow. Rate of flow is readily controlled by changing the speed of the rotating plate and adjusting the height of the delivery funnel above the surface of the dust contained in the dished plate. A scraper fixed over the surface of the rotating plate forces the dust into a discharge tube positioned above the test cleaner, see Fig. 18. The efficiency determination is based on the difference in weight of the air cleaner before and after test, the amount of test dust fed into the cleaner being accounted for by weight. For this purpose a Bullion type precision balance is used. It is capable of weighing loads up to 75 kilogrammes to an accuracy of 0·50 grammes, that is, up to 1 in 150,000 to enable the required degree of small differences to be assessed.

It is appreciated that the direct method of dust feed is not truly representative of operating conditions in the field. However, this method has been given preference for test purposes over the use of an artificially produced dust storm owing to difficulties encountered during tests conducted in the dust storm cabinet in aspirating the heavier dust particles in cloud formation.

In the case of the oil bath air cleaners, it is necessary to ensure that oil is not carried over to the outlet under any circumstances, not even when the cleaner is working in a 30 deg forward or backward 15 deg sideways tilted position. A glass panel is inserted in the outlet side of the cleaner and it is possible to detect oil carry-over by examining this for the presence of oil particles.

Shock test machine

This machine was installed as a means of assessing and analysing damage or operational failure which may occur to equipment, mainly electrical and optical, attached to or in close proximity to bulkheads or turrets, vulnerable and subject to sudden blow caused by the impact of missiles or the blast of adjacent explosions. The machine consists of a steel framed structure with a sprung anvil plate which can be mounted in three positions to receive end, vertical and horizontal impacts up to approximately 1 ft/ton in intensity applied by a 420 lb, weighted pendulum or an ingot falling from controlled heights. Test units are

A. Tyres mounted for testing B. Driving drum
Fig. 13. 10-ton tyre testing machine

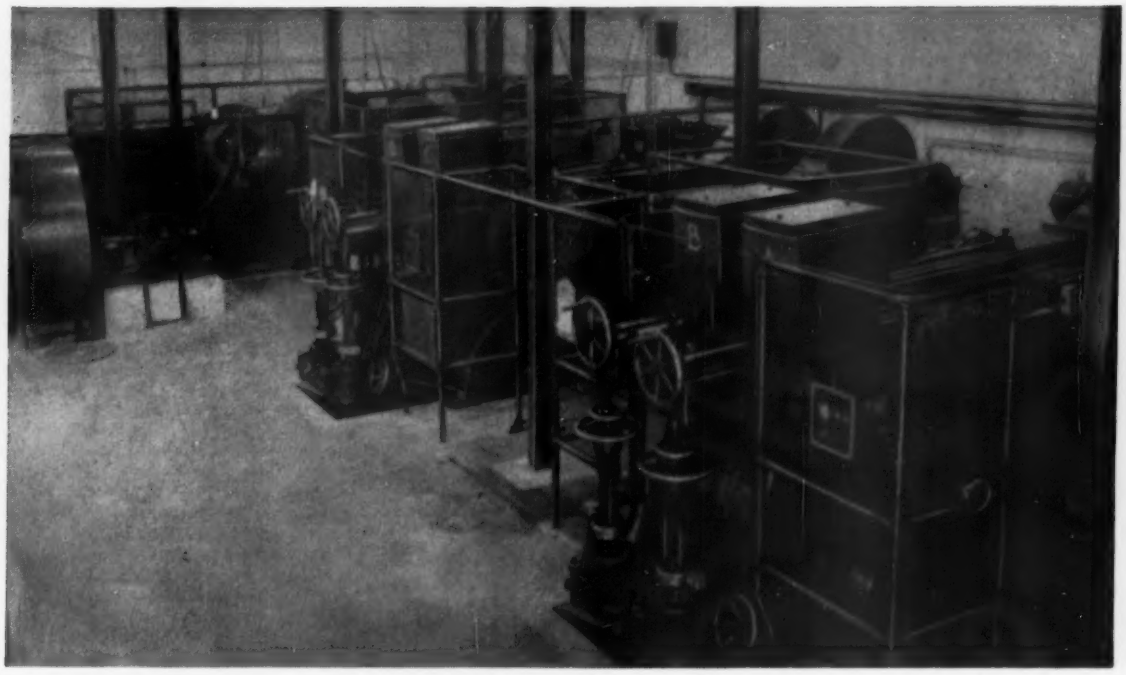


Fig. 14. General view of tyre and wheel testing laboratory

mounted on the sprung anvil plate. The test procedure is to apply a series of blows of gradually increasing intensity according to a test schedule based on empirical information obtained over a considerable period of testing and consolidated in some degree by experience of field service. The consistency of the performance of the machine was repeatedly checked at frequent intervals by crusher gauges. Although these gauges do not give quantitative calibration they are indicative of the consistent performance of the machine for shock test comparative assessment. This equipment is shown in Fig. 19.

Moving coil vibrator and vibration table

To test the ability of fighting vehicles and component assemblies to withstand vibrations and to carry out exploratory resonance tests a moving coil type resonator and a vibration table are employed. The moving coil vibrator test gear is for general purpose vibration excitation of medium and large components and structures at frequencies from 20 to 500 cycles per second. It may be used for the investigation of resonant frequencies and modes of vibration in fighting vehicles of all sizes, and can also be used for vibration endurance, functioning and fatigue testing of vehicle-borne equipment and apparatus.

The equipment is made up of a Ward-Leonard type motor generator set which can be used to control the speed of either of two motor alternator sets, covering the low frequency and high frequency part of the test spectrum respectively. The alternators are coupled through an impedance matching auto-transformer to a large moving coil vibrator fitted with an air-blast cooling blower. A control cubicle is provided for speed control and monitoring purposes, and a moving coil vibration pick-up and a high accuracy cathode ray oscilloscope are supplied for the measurement of vibration amplitudes and phase-relationships, see Fig. 20. A cam operated table to reproduce certain accelerations and amplitudes met with in vehicles was used to subject small assemblies to test. The table, 30 in², can be adjusted to give amplitudes up to 0·10 in at frequencies up to 70 cycles per second, see Fig. 21.

A. Adjustable length impact spike B. Drop table
C. Recorder D. Release mechanism

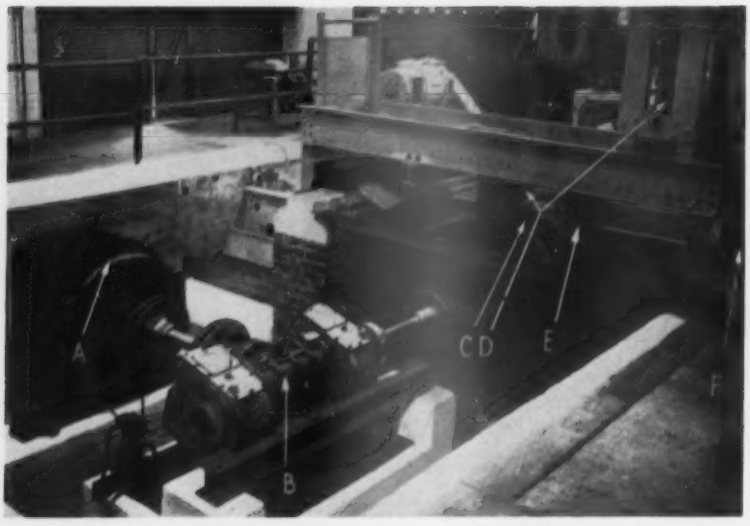
Fig. 15. Tyre impact machine

Burst test spindle

In order to subject rotating parts to the centrifugal forces experienced in automotive high speed work a rigid spindle with a suitable face plate or chuck was horizontally mounted in pedestal bearings and installed in a totally enclosed timber-lined concrete pit so that bursting tests can be carried out safely. The spindle was coupled with a propeller shaft through a hole in a thick steel protection shutter to a 20 h.p. electric variable speed motor and driven at speeds up to 9,000 r.p.m. A number of tests have been made on flywheels, clutches, fans and similar components and by the use of a flashing lamp stroboscope and a high speed camera, useful information has been obtained on the behaviour of components at speeds sufficient to cause distortion and on occasions total disintegration.

Torsion bar fatigue test machine

A torsion bar fatigue testing machine to subject torsion bars to angular twist and maximum torque consists of a 15 h.p. motor driving a five-speed gearbox and multiple belt reduction gear to a large flywheel at 65-80 r.p.m. It is shown in Fig. 22. The flywheel acts as a pulley and torque smoother. Through a linkage, an adjustable throw crank with adjusting screw and scale on the flywheel shaft, oscillates a



A. 40 h.p. variable speed motor B. "Covenanter" gearbox C. Uneven cam
D. Resilient wheel and suspension on test E. 4 ft diameter drums F. Detachable ramp

Fig. 16. Rig for testing tank hulls, chassis, suspension units and wheels under dynamic loading

second crank which in turn actuates the two headstock levers for applying the required angular torque to the torsion bars under test. The oscillating crank is provided with a centre point lock, used when the movement of the headstock levers is being adjusted and also for equalizing the strain on the torsion bars during temporary cessation of tests.

Quadrant scales on the headstock indicates the angular movement of the headstock lever and a rev-counter driven from the oscillating crank records the number of oscillations. This rig is shown in Fig. 23. Safety covers are provided to encase the torsion bars on test in case of a break and failure. The rig is capable of testing bars up to 2·45 in effective diameter and 10 ft long to a maximum effective diameter of 140,000 lb/in² at a frequency of 70 oscillations per minute with a maximum angular twist of 46 deg.

Final drive and transmission test equipment

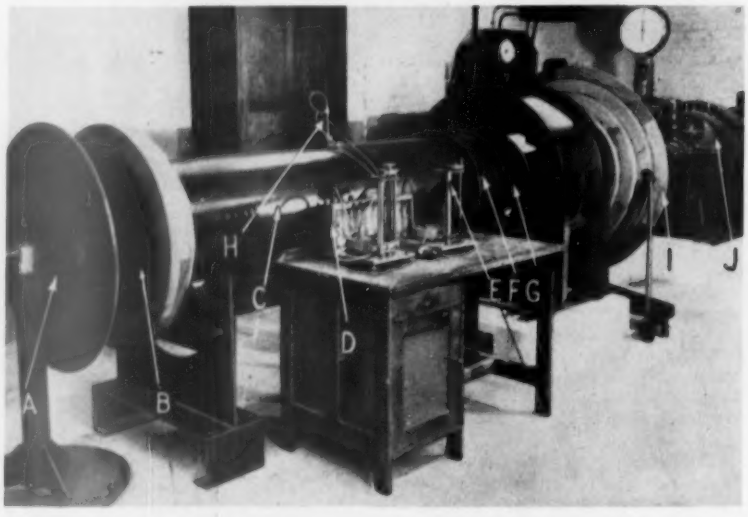
For testing gearboxes, torque convertors, coupling and transmission components, an electrical test rig is provided, see Fig. 24. This plant comprises swinging field motoring and absorption dynamometers with Ward-Leonard controls. The system is a regenerative one, the power from the output being fed back electrically to the input, so that only the power losses need be supplied from the mains. The input power varies from 130 h.p. at 2,000 r.p.m. to 200 h.p. at 4,000 r.p.m. and the whole system is reversible both in power flow and direction of rotation. A special feature is the high output torque which can be dealt with at output speeds down to 250 r.p.m. For testing final drive gears a rig, see Fig. 25, was devised by placing two assemblies back-to-back and imposing a torque between them, thus it was necessary to provide only the power losses while testing these assemblies under full load. Two final drives are positioned on either side of a bed plate and the input and output drives are connected by shafts. One of the shafts is resilient, being in fact a tank suspension torsion bar. When setting up for test, a vernier flange coupling on the resilient shaft is disconnected and one set of gears locked solid. A known torque is applied to the resilient shaft by rotating one side of the flange coupling in relation to the other. The coupling is re-made and locked in the set load position so that when the rig is running and wear does not occur in the teeth or bearings of the gears, the imposed load will remain constant. To enable the torque to be checked during the test while the rig is running, a strain gauge attached to the resilient bar is connected to a slip ring pick-up. Whilst a pinion shaft torque of up to 100,000 lb-in may be locked in the system, a 25 h.p. motor is capable of driving the rig and up to 450 b.h.p. can be passed through the gears. Cooling and temperature control of the oil in the gearboxes was catered for in the development of the rig and has proved successful.

Equipment for investigation of welding properties of armour plate and electrodes

For investigating the welding characteristics of ferrous materials a static transformer arc welding set is used. The welding power supply equipment comprises a 34 K.V.A. continuously rated oil-cooled transformer having a capacity of 600 amps at 100 volts, together with two regulators, one of which gives infinite adjustment between 30 and 450 amperes. For the cutting of plate before welding a Circaline oxy-acetylene gas cutting machine is provided. This machine is motor driven and has an adjustable cutting head which enables cuts to be made at angles up to 90 deg from the vertical plane. A 20 in diameter abrasive wheel cutting-off machine is used for cutting out specimens after welding. This machine is of the dry cutting type and is fitted with adjustable tables and driven by a 20 h.p. motor. For the preliminary preparation of the specimens after cutting off, a heavy duty linishing machine is used. A bench type metallurgical moulding press with necessary attachments was available for mounting specimens to be microscopically examined.

Tropical test facilities

The tropical test facilities include provision for subjecting complete



A. Throttle plate on adjustable stand B. Air outlet C. Main tunnel D. Sloping manometer
E. Micro manometer F. Air straightener G. A section of the main tunnel has been removed to show fan
H. Pitot-static tube J. Motoring dynamometer

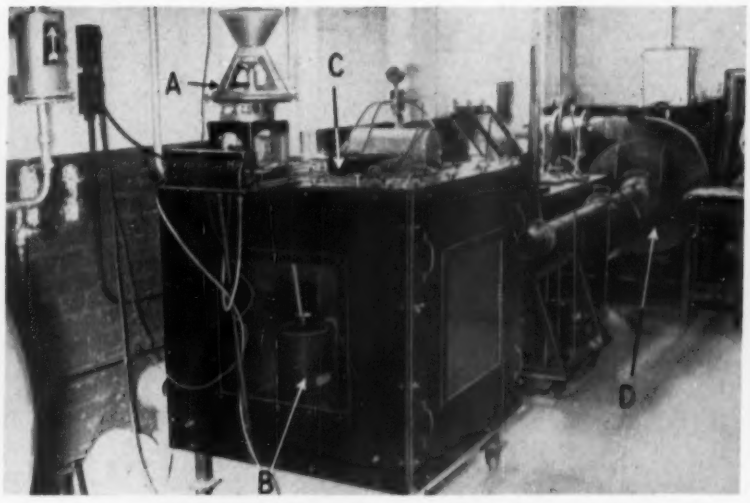
Fig. 17. Cooling fan test rig

armoured vehicles or bulky equipment to moderately high temperatures with very high relative humidities, or extremely high temperatures with low humidity, thus simulating either extreme jungle or desert conditions.

Essentially this chamber, consists of of an insulated sheet steel cell 36 ft × 20 ft × 16 ft high, erected inside a laboratory, the cell being heated internally by means of steam pipe coils and solar ray lamps. The cell is maintained at any required temperature level; this ensures extremely steady conditions independent of outside weather effects. The humidity of the air in the cell is controlled by injecting low-pressure steam from sparge pipes, and the effect of intense sunshine can be simulated with batteries of solar ray lamps suspended from the ceiling. Temperatures up to 50 deg C at 100 per cent relative humidity, or high temperatures up to 93 deg C with lower humidities, can be maintained over indefinite periods.

A specified method of operation is to keep temperatures and humidity high during the day and allow the system to cool naturally during the night, repeating this cycle for indefinite periods to allow the breathing effects on equipment under test to be assessed and investigated. For some tests fungi spores are sprayed on the equipment to study the efficacy of tropic protection measures from the fungi growth and attack aspect. Relative humidities and temperatures within the cell are measured by remote indicating wet and dry thermo-couples over which air is aspirated by a small fan. Instrument measurements on equipment under test are recorded in a laboratory adjoining the test cell.

This instrumentation includes a 100 point thermo-couple recorder. A tunnel with entry doors and observation windows into the tropical cell runs the full length of the chamber and the temperature is controlled to protect personnel leaving the cell from exposure to sudden temperature changes. The tunnel leads to hot and cold shower baths, warm dressing and rest rooms, and the instrument recording and control rooms.

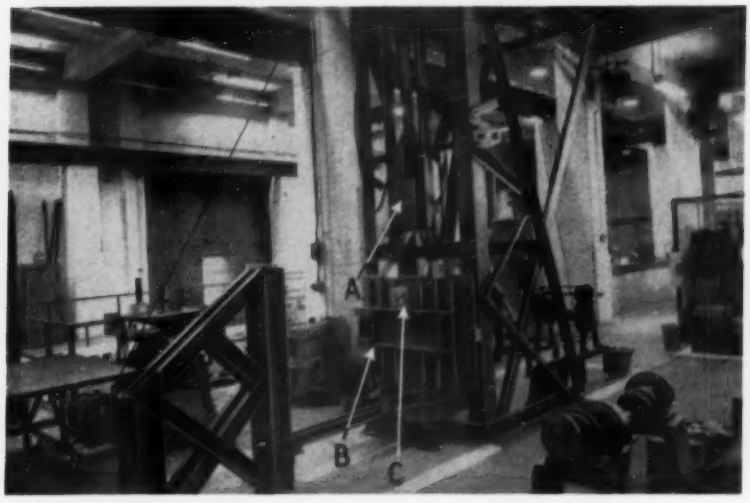


A. Dust feed mechanism B. Cleaner on test C. Air inlet orifice D. Variable speed exhauster fan

Fig. 18. Test rig for fighting vehicle air cleaner

The tropical test cell has been in continuous use since its installation for tropical climatic investigations and development of engines (both for land and air use) armoured tanks, radar equipment, electrical installations, optical instruments, packing and tropic proofing materials, fire control equipment, ambulance, office photographic and X-ray wheeled vehicles.

To test and assess the performance of small components under the hot and dusty conditions obtaining in desert operations, such assemblies as fans, dynamos, generators, distributors, and radiators, are mounted in a chamber 4 ft² × 4 ft high and operated under controlled conditions for any desired test endurance period. Erosion effects are obtained by continuously feeding dust into the chamber at a point where the pressure is below atmospheric. This technique has been found to reproduce closely, the results of field trials, with the advantages of continuous observations, minimum of instrumentation and a considerable saving of time.



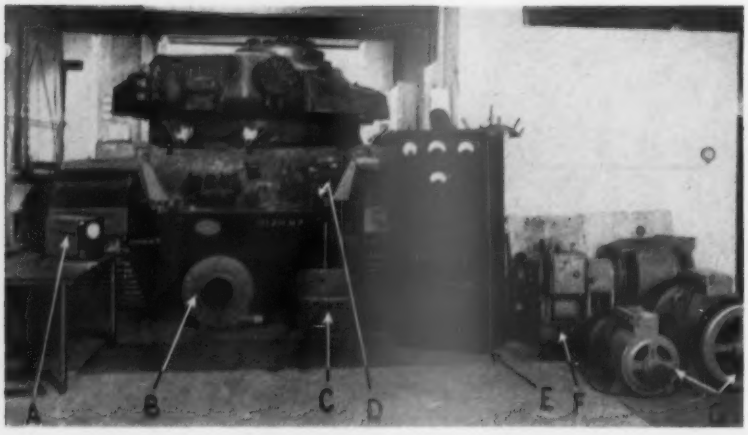
A. Drop weight B. Sprung anvil plate C. Electrical equipment under test

Fig. 19. Shock test machine at F.V.R.D.E.

Low temperature test facilities

The low temperature test facilities available at F.V.D.R.E. are comprehensive and consist of:—

(a) A laboratory with a large chamber 32 ft square and 16 ft high, capable of operating at temperatures down to —57 deg C. The test chamber is cooled by circulating air in a closed circuit alternately through the chamber and a remote evaporator. Refrigeration is provided by an ammonia absorption plant working on the Maiuri system, in which ammonia is alternately evaporated from, and reabsorbed in an aqueous ammonia solution. The heat necessary for the evaporation is supplied by steam coils connected to the Establishment and the works general heating system; the energy derived from the steam takes the place of that supplied by the compressor motor in a mechanical refrigeration system. This equipment is one of the largest absorption plants in existence and, aided by a pre-cooled brine circuit used in conjunction with the condenser, can deal with steady heat inputs up to 600,000 B.Th.U. per hour, maximum loads of 750,000 B.Th.U. per hour, for periods up to 30 min. For reheating the chamber after a cold period, a battery of gilled steam coils with a capacity of 1,500,000 B.Th.U. per hour is provided in the air circulating system. An electrical dynamometer is provided capable of absorbing 175 h.p. or delivering 95 h.p. at speeds up to 3,000 r.p.m. for measuring torque, friction losses, driving gearboxes and investigating engine starting and operating characteristics.



A. High accuracy cathode ray oscillograph B. Air blast cooling blower C. Moving coil vibrator
D. Moving coil vibration pick-up E. Control cubicle F. Ward-Leonard set G. Two motor alternator sets

Fig. 20. Set-up for testing ability to withstand vibrations

When an engine is run in the test chamber, combustion make-up air is drawn from the atmosphere through a dryer and the exhaust gas is passed direct to the open air outside the building. The laboratory is equipped so that temperature and instrument measurements within the test chamber, and on equipment being tested, are recorded in a control room adjoining the low temperature chamber. Safety precautions include a CO_2 system of fire extinguishers, apparatus for continuous sampling of the air in the test-chamber for any accumulation and build-up of toxic fumes from equipment under test, and two-way conversation arrangements between the test chamber and the laboratory control room. Special clothing and equipment with body heating facilities and life saving equipment are provided. The author is preparing a scheme for an extension to the low temperature facilities to enable storm and icing conditions to be simulated.

(b) A small refrigerating cabinet approximately 3 ft cube giving controlled working temperatures down to —45 deg C and provided with an external electric motor for driving components under test is used for investigations and performance at low temperatures of small specimens and equipment.

Test equipment for general application

Motoring dynamometers for general use

The test equipment includes three swinging field electric motoring dynamometers of 10, 20 and 40 h.p. capacity respectively at speeds up to 1,500 r.p.m. These can be readily moved with their control gear for incorporation in any test rig for which they are required. They are of the alternating current type driving through magnetic couplings, the output speed being varied by adjusting a small D.C. current which is passed through a control winding on the coupling. Owing to a certain degree of speed instability with temperature rise, this system has not been found entirely satisfactory.

Hounsfield tensometer

This portable tensile and compressive testing machine for loads up to 2-tons can also be used for bending and hardness tests.

Pendulum impact machine

The Hounsfield balanced impact testing machine uses a beam type notched test piece, and to increase the speed of impact, both hammer and anvil (holding the test piece) are arranged as pendulums travelling in opposite directions.

Vickers hardness machine

This machine is of the standard type, in which diamond pyramid or ball hardness tests can be carried out.

Deadweight pressure gauge test equipment

To give an independent check on the accuracy of pressure gauges, these are calibrated on a Bourden deadweight rig in which pistons of known diameter are loaded by standard weights and the pressure transmitted by an oil system to the gauge under test.

A. Variable speed control B. Driving motor
C. Optical amplitude gauge

Fig. 21. Cam-operated table for vibration tests on small assemblies

Thermometer calibrating baths

The equipment includes a series of temperature controlled baths of water, oil and salt in which distant reading or other thermometers and thermocouples can be checked against standard mercury-in-glass thermometers.

Vickers projection microscope

Although primarily installed for metallurgical work, particularly the examination of weld sections, the Vickers projection microscope can be used for visual or photographic examination of any small specimen by transmitted or reflected light, using colour filters as necessary. The greatest magnification for which equipment is available on the laboratory instrument (type M500) is 4000 diameters.

Tensile testing machine

A self indicating universal tensile testing machine having a maximum capacity of 50 tons, reading by 0·050 ton divisions, for use on materials testing and investigations.

High capacity sensitive balance

The precision balance installed in the test laboratories is capable of weighing loads up to 75 kg to an accuracy of 0·10 gramme. Its chief use is in investigations on air cleaner performance tests, where the method used depends on accurately determining differences between the weight of a cleaner before and after test.

Miscellaneous and laboratory instruments

Apart from normal laboratory instruments for the measurement of temperature, vacuum, pressure, speed and electrical characteristics, the following special items may be mentioned.

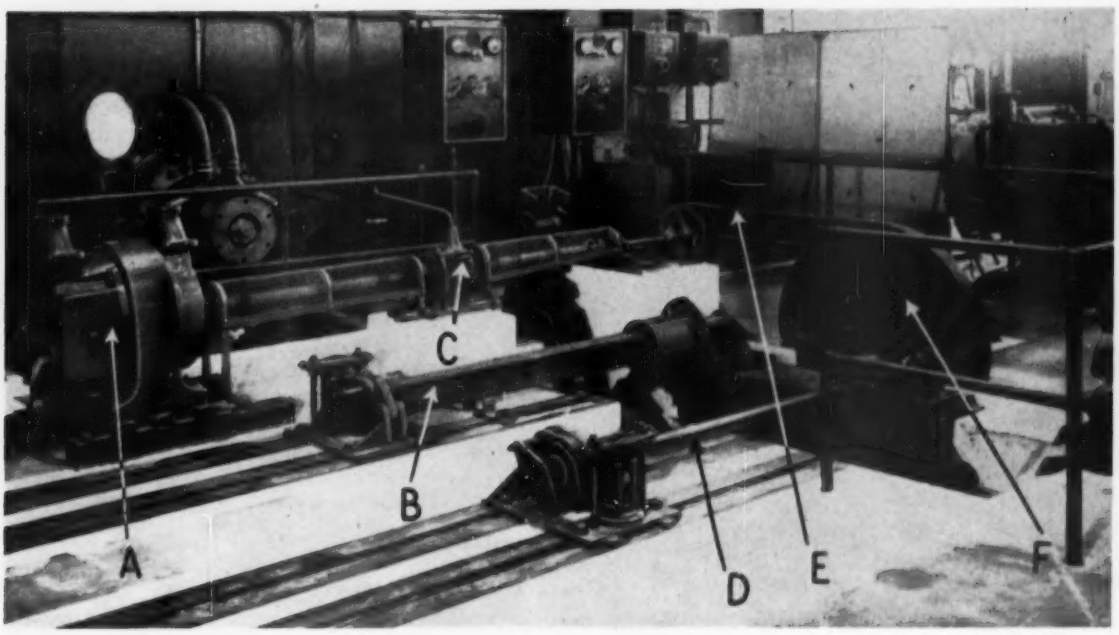
N.P.L. pattern objective noise meter.
Standard telephone Company subjective noise meter.
Cambridge exhaust gas analyser.
Vane anemometers, measuring speeds up to 10,000 ft/min.
"Metrovick" Velometer for air flow measurements.
"Stroboflash" flashing lamp stroboscope.
"Andreason" sedimentation pipette.
Everett Edgcombe photometers of various ranges.
Oscillographs for use with electrical strain gauges and vibration pick-ups, fitted with continuous film cameras for recording.
Foxborough indicators and dial recorders for resistance strain gauges.
Induction and piezo-electric accelerometers.

Specialized test rigs

A large number of special test rigs have been built from time to time to cover the requirements of particular tests. These have included the following:

(i) A "mud bath" for comparative suspension bearing wear tests. A tank road wheel was immersed in wet mud to above axle level and driven by means of a roller under a controlled load.

(ii) An oil seal test rig, in which four shafts fitted with oil seals are arranged



A. 15 h.p. motor B. Safety cover C. 3-speed gearbox D. Exposed torsion bar E. Multiple belt drive F. Flywheel

Fig. 22. Torsion bar test equipment

so that the seals can be subjected to oil pressure on one side and muddy water on the other.

(iii) A tensile test frame for lengths of tank track, operated hydraulically and capable of applying loads up to 75 tons.

(iv) A model air flow analyser, consisting of a glass topped chamber in which can be incorporated profiles simulating those of cooling air etc., passages, visual observation of the air flow through the chamber being made possible by the injection of smoke.

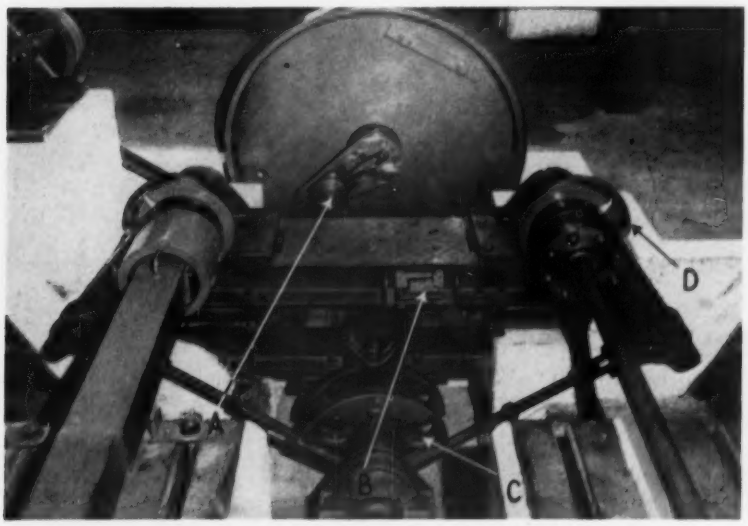
(v) Rigs for studying oil circulation in main gearboxes and final drive units. In these investigations the unit for test is mounted and run in similar conditions as in the vehicle, with celluloid windows installed to allow visual or photographic study of the oil circulation. Power absorption is measured by driving with a motoring dynamometer.

(vi) A steam pressure rig for radiator relief valve testing. The relief valve under test was connected to the steam mains through an orifice flow meter and the pressure controlled by a reducing valve. The throughput of the valves is measured as a function of pressure.

(vii) An endurance test rig for hydraulic turret traverse pumps. The pump was connected in a closed oil circuit and the discharge restriction was varied according to a regular cycle by using a control valve governed by a pneumatic servo mechanism.

(viii) A rig for life tests on cooling fan driving belts. In this case a fan was driven from an engine through a clutch and the belts under test, and arranged so as to force air through a duct provided with a damper reproducing the same restriction as met with in the vehicle installation. Repeated loads were applied to the belts by successive engagements of the clutch, to simulate the fatigue effect of sudden engine acceleration met with in service, particularly when changing gears (double declutching).

(ix) Engine cooling water pump tests have been carried out by connecting the pump in series with a calibrated venturi or inferential water meter, in a closed circuit. The water is prevented from becoming unduly hot by including a water-cooled heat exchanger in the circuit. A more satisfactory method is, however, to use calibrated measuring tanks, and a permanent rig of this type is under construction.



A. Linkage and adjustable throw crank B. Oscillation counter C. Oscillating crank D. Quadrant scale

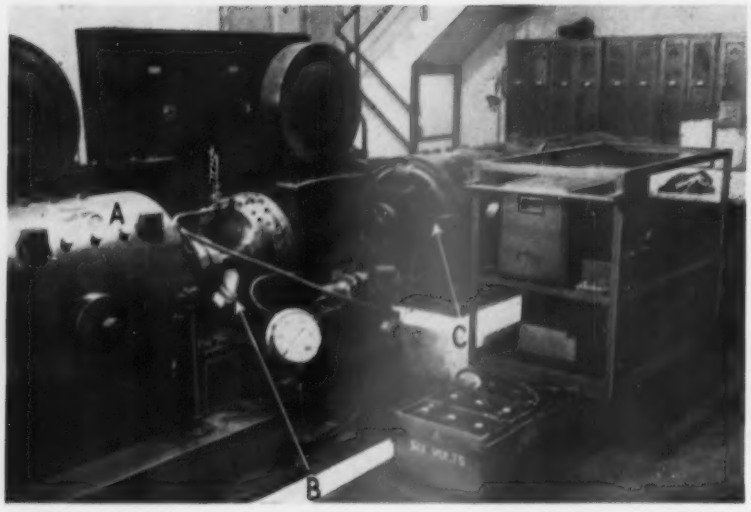
Fig. 23. The mechanism for applying torque to torsion bars

Summary and conclusions

The information given in the present paper may be summarized as below:—

Origin of the Engineering Research Test Laboratories

It was realized early in the war that considerable time and expense could be saved in the preliminary stages of development of fighting vehicles and equipment by the use of rig test methods, and a test house was established at the Experimental Wing, Farnborough, late in 1941. Results of



A. Motoring dynamometer B. Disc brake on test C. Absorption dynamometer

Fig. 24. Electrical rig for testing gearboxes, torque converters, coupling and transmission components

considerable value to guide future design of armoured fighting vehicles and equipment were obtained expeditiously even with the crude equipment available, particularly in respect of the engine cooling performance of vehicles under full power conditons.

The present laboratories, completed on the Longcross site during the war, embody much of the experience gained previously. Attention was at first necessarily devoted solely to *ad hoc* problems, but the equipment was designed with the requirements of future engineering research in mind. The author of this paper proposed a scheme and submitted his plans in 1947 for a new building and extension to the laboratories to meet the extending demands being made upon the facilities available. The suggested extension received Ministry approval and financial sanction. The new building is in course of erection and additional equipment is on order. At present (May 1955) the staff consists of: A Superintendent Chief Experimental Officer, two senior Experimental Officers, two Experimental Officers, two Engineers, six Technical Officers, two Scientific Assistants, three Design and Drawing Office staff, two typing and clerical assistants and 45 industrial staff comprising: research master mechanics, machinists, fitters, labourers and storekeepers.

The following general conclusion may be drawn from some fourteen years experience of rig test work on armoured fighting vehicles and equipment design:—

(i) Even though in some cases crude methods were necessary because of lack of facilities, equipment and time, it has been possible to simulate user conditions for most of the rig test work and engineering research investigations for almost all the conditions that the vehicles and equipment are likely to meet in actual service.

(ii) Where possible it has always been arranged to confirm the results of laboratory rig tests by operational experience. Laboratory tests have proved valuable for purposes of prototype testing and have made possible considerable saving in costs, time and labour. In certain circumstances rigs and test equipment have provided a degree of control over test conditions which it would have been impossible to obtain in the field. This is particularly so in the case of air cleaner development tests, development and testing of cooling system fans, engine cooling trials on complete vehicles, and performance tests of vehicles and equipment at extreme climatic temperatures.

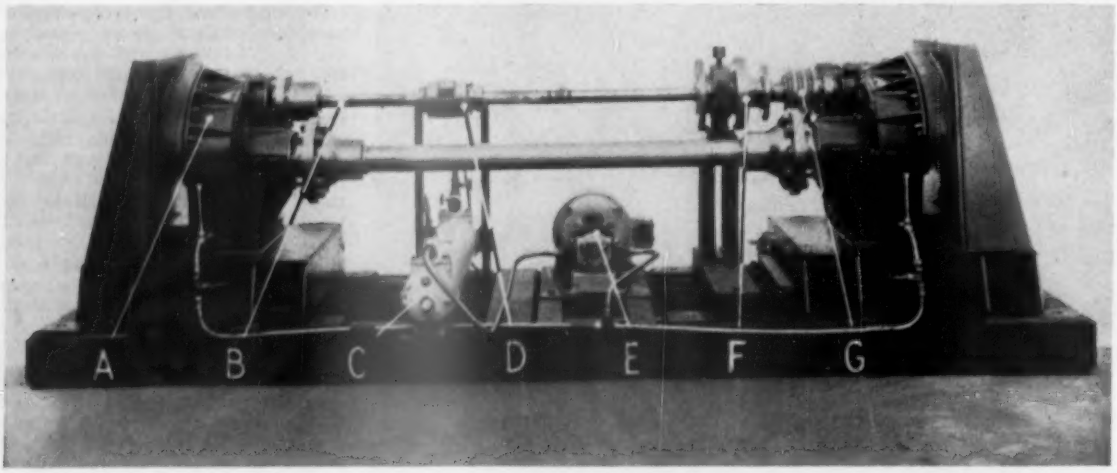
(iii) The decision to establish the Engineering Research Test Laboratories has been fully justified by the resulting benefits to designers and users.

The 1,000 or more reports which have been issued on a wide variety of projects since the establishment of the Engineering Test Laboratories give some indication of the scope and activities of the work of the organization during the comparatively short time of its existence.

Acknowledgments

The author wishes to record his grateful appreciation of the assistance rendered by his colleagues, particularly Mr. H. J. Purkiss and Mr. N. T. Leach, who so ably carried out the design and development of the rigs and equipment. The illustrations are Crown copyright reserved, and are reproduced by permission of the Controller, H.M. Stationery Office.

Acknowledgment is made to the Chief Scientist, Ministry of Supply, for permission to publish this paper.



A. Final drives B. Resilient shaft C. Oil cooler D. Strain gauge and slip ring pick-up E. Motor and oil pump F. Vernier shaft coupling G. Sprocket for chain drive

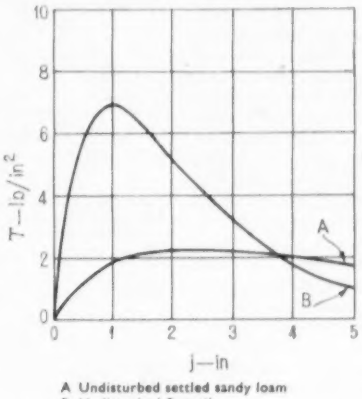
Fig. 25. The testing equipment for final drives

WHEELS OR TRACKS

Traction and Slippage of Wheels and Caterpillar Tracks on Soft Terrain

Lt.-Col. M. G. Bekker, Dipl. Ing.

IT is, of course, well known that even for the same unit loads a crawler vehicle may cross soft terrain without considerable slippage whereas the wheels of road vehicles may slip excessively, or spin. The amount of slip varies with soil type. This has caused considerable controversy regarding the extent to which wheels, particularly with low pressure pneumatic tyres, may replace tracks or *vice versa*. However, from the relationship between certain physical characteristics of soil and its shearing strength, a rational solution can be obtained. Since, from the point of view of mechanics, both a wheel and track may be considered simply as loading surfaces having similar forms but different lengths and widths, some data determined for a crawler track may be assumed to be valid also for a wheel, provided the loads and the geometry of the application are properly considered.



A Undisturbed settled sandy loam
B Undisturbed firm silt

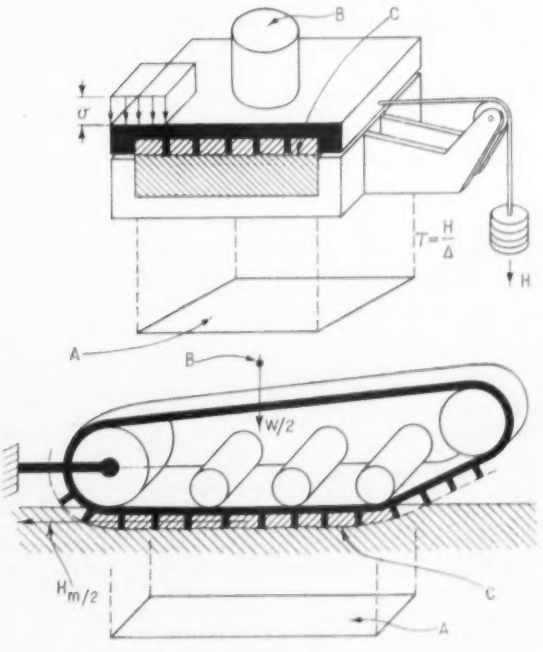
Fig. 3. Shear stress plotted against slip

The track

The general significance of the analogy between the soil shear produced by laboratory apparatus and the shear caused by a track, as shown in Fig. 1, requires explanation. If the upper half of the shear box, or the blocked track, is moved with reference to the lower half of the soil mass, the maximum shearing force is not developed instantaneously with the start of the relative motion: this is because the soil must be compacted to a certain degree. Thus some slip takes place initially before the full shearing stress is developed. Assuming that the ground is homogeneous, all the grousers encounter the same resistance and slip to the same extent, during the shearing process.

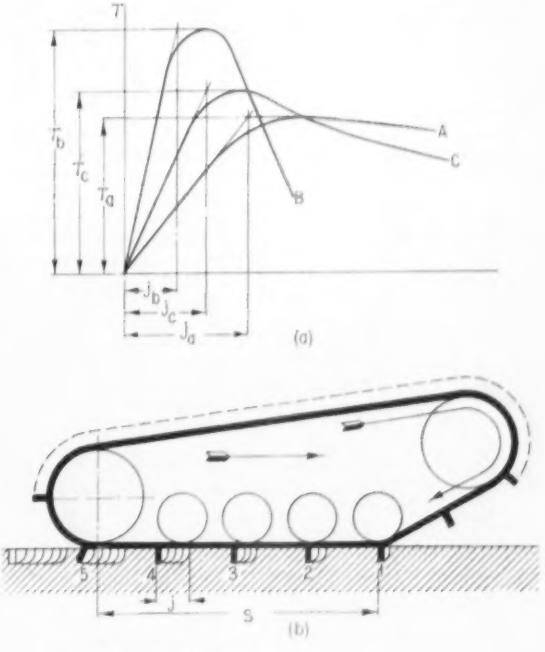
If the relative motion of the soil under shear is plotted as a function of the corresponding stress, three basic curves can be determined; their shapes will depend on the three types of soil strength involved. Fig. 2a shows the general character of the curves. Curve A is for loose frictional or plastic soils. The shearing strength τ_a of such a soil is reached after the initial period of compaction, which takes place over a distance j_a. Then the stress remains practically at the same level, irrespective of any further slip, d.

Soil B is of different structure; it is not loose but forms a more or less solid coherent mass. Dry, settled clays or firm silts in the undisturbed state may belong to this category. Snow at very low temperatures has similar characteristics. At these low temperatures, snow more closely resembles a spongy cohesive mass than a loose granular structure. This type of material quickly reaches the required degree of compression, j_b, and then shears off rapidly. In this process,



A Approximate shear area △
B Load
C Approximate line of shear

Fig. 1. Soil shear analogy



A Loose frictional or plastic soils
B Dry settled clay or firm silt
C Soil with intermediate properties, that is, between those of A and B

Fig. 2. The characteristics of soil deformation

the original structure is destroyed; the new structure does not have the strength of the original mass, and the τ_s value decreases rapidly.

The third type of soil, *C*, has intermediate properties: upon reaching the required compaction at the slip distance j_c, which corresponds to the maximum value of τ_c, the shearing strength also decreases, but not as rapidly or to such a low value as in Case *B*. Average agricultural soils and most types of snow at temperatures approaching melting point belong to this category. These phenomena take place when the whole surface of the body is in uninterrupted contact with the soil during the shear process. This condition may be assumed to be satisfied if the shear is performed in accordance with the scheme shown in Fig. 1, that is, when the track is blocked before it is pulled relative to the ground.

In the case of a vehicle in motion, however, the conditions are somewhat different. Fig. 2b shows the front portion of the track, adjacent to the first grouser, coming into contact with the undisturbed soil. At this point, the shearing force cannot be the same as at the other grousers, 2, 3, 4 and 5, where the ground has been loaded for a certain period of time. It is only after the front portion of the track has moved back, relative to the vehicle, in contact with the soil that the gradual compression of the soil, or slipping of the track on the soil surface, enables the full tractive effort to be produced. The maximum slip is reached before section 1 of the track leaves the ground at the rear of the vehicle. Since the track cannot stretch, the amount of horizontal coil distortion or slip, j, that occurs beneath each of the track sections 1, 2, 3, 4 and 5, increases progressively along the ground contact area, Fig. 2b. Thus, it can be seen that a stalled track, as shown in Fig. 1, does not accurately reproduce all the effects of a crawler in motion. This fact, although stressed a long time ago, is seldom fully appreciated.

It is an interesting coincidence that the form of the soil shear curves shown in Fig. 2a is identical to the form of the curves pertaining to the displacement, x, and natural frequency time, $\omega_n t$, of an aperiodic vibration. The general equation of such a motion is:

$$x = A_1 e^{(-b+\sqrt{b^2-1})\omega_n t} + A_2 e^{(-b-\sqrt{b^2-1})\omega_n t}$$

where b is the coefficient of damping, which is larger than unity. To write a corresponding equation in terms of soil deformation j to replace $\omega_n t$, and of soil stress τ substituted

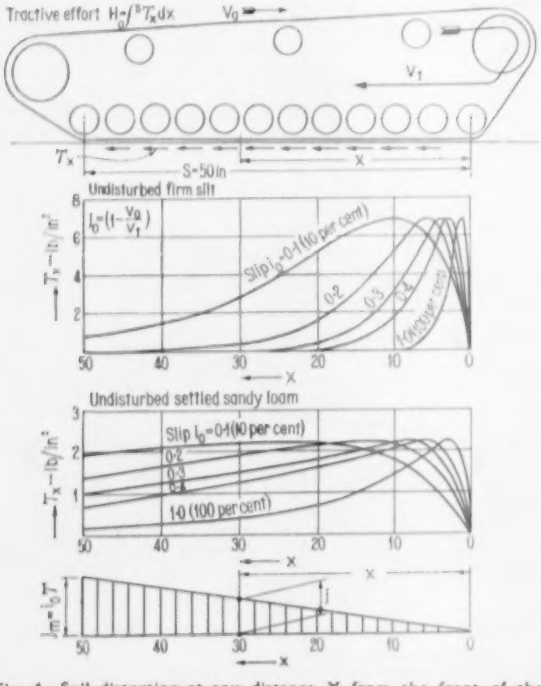
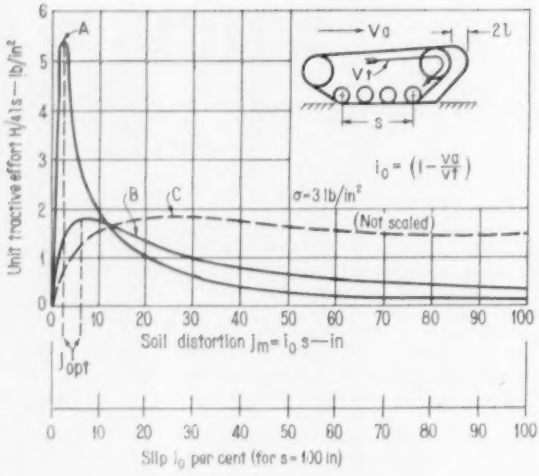


Fig. 4. Soil distortion at any distance X from the front of the ground contact area



A Undisturbed firm silt (C=6·5 lb/in², ϕ=10 deg, K_1=1, K_2=1·1)
B Undisturbed settled sandy loam (C=0·2 lb/in², ϕ=35 deg, K_1=0·3, K_2=2)
C Uniform plastic or loose frictional soil ($K_1 \approx 0·2$, $3 < K_2 < 4$)

Fig. 5. Variation of tractive effort per unit track area with soil distortion

for the displacement x, it is necessary to introduce the coefficient K_1, from which the scale of j with reference to τ can be determined. This coefficient depends on the degree of soil looseness or compactness and reflects the amount of slip, j, required to obtain the corresponding value of τ. Furthermore, the type of curve is determined by the coefficient, K_2, which replaces the value of damping, b, and predetermines whether the shear curve is of the *A*, *B*, or *C* type, Fig. 2a. Finally, to evaluate the constants A_1 and A_2, the coefficient K_3 is introduced; this is defined as the slope of the tangent to the curve at the beginning of the co-ordinates. According to these assumptions,

$$\tau = A_1 e^{(-K_2+\sqrt{K_2^2-1})K_1 j} + A_2 e^{(-K_2-\sqrt{K_2^2-1})K_1 j} \quad \ldots\ldots(1)$$

The value K_2, as has already been mentioned, is to be included in the constants A_1 and A_2. These constants can be determined in the following way: For slip $j = 0$, $\tau = 0$; hence,

$$0 = A_1 + A_2 \quad \ldots\ldots(2)$$

Also for $\tau = 0$ and $j = 0$, $d\tau/dj = K_3$, and:

$$\frac{d\tau}{dj} = A_1 e^{(-K_2+\sqrt{K_2^2-1})K_1 j}(-K_2+\sqrt{K_2^2-1})K_1 + A_2 e^{(-K_2-\sqrt{K_2^2-1})K_1 j}(-K_2-\sqrt{K_2^2-1})K_1 = K_3$$

or,

$$K_1(A_1 - A_2)\sqrt{K_2^2-1} - K_1 K_2 (A_1 + A_2) = K_3 \quad \ldots\ldots(3)$$

From equations (2) and (3),

$$A_1 = \frac{K_3}{2K_1\sqrt{K_2^2-1}}$$

$$A_2 = -\frac{K_3}{2K_1\sqrt{K_2^2-1}}$$

After substituting these values into equation (1), it will be found that:

$$\tau = \frac{K_3}{2K_1\sqrt{K_2^2-1}}\left[e^{(-K_2+\sqrt{K_2^2-1})K_1 j} - e^{(-K_2-\sqrt{K_2^2-1})K_1 j}\right] \quad \ldots\ldots(4)$$

The coefficient K_1 reflects the degree of soil compactness, in accordance with the original assumption. Its value can be determined, for instance, by trial and error when plotting the τ (j) curve. It is believed that the value of $K_1 = 0·3$, is representative of an average type of soil. A brittle, or highly coherent, soil may have a value of K_1 close to unity. The coefficient K_2, which determines the fundamental character of the shear curve, apparently reaches a value of 2 for an

average soil and approaches unity for brittle soils; it is always larger than unity. Loose, dry sands or supersaturated ideally plastic clays may have a value of K_2 approximately equal to 3. The coefficient K_2 is dimensionless and its value can be established empirically from the stress-strain curve obtained from a shear test.

Since, in equation (4), the portion enclosed by the brackets is dimensionless, the value of $K_3/2K_1\sqrt{K_2^2-1}$ must have the dimension of lb/in², so τ is expressed in the same units. The value of K_3 may be defined in the following way. From the equation based on the Mohr-Coulomb hypothesis of soil failure, it follows that $\tau_{max}=c+\sigma \tan \phi$. Thus, the maximum value of the coefficient enclosed by the brackets of equation (4) multiplied by $K_3/2K_1\sqrt{K_2^2-1}$ is equal to $c+\sigma \tan \phi$.

Hence,

$$K_3=\frac{2K_1\sqrt{K_2^2-1}\,(c+\sigma \tan \phi)}{\left[e^{(-K_2+\sqrt{K_2^2-1})K_1 j}-e^{(-K_2-\sqrt{K_2^2-1})K_1 j}\right]_{max}} \quad \ldots\ldots\ldots(5)$$

Now, equation (4) can be written in a simplified form if the maximum value of the function enclosed in the brackets of the denominator of equation (5) is denoted by y_{max}:

$$\tau=\frac{c+\sigma \tan \phi}{y_{max}}\left[e^{(-K_2+\sqrt{K_2^2-1})K_1 j}-e^{(-K_2-\sqrt{K_2^2-1})K_1 j}\right] \quad \ldots\ldots\ldots(6)$$

Thus, the constant K_3 has been replaced by a function of the stress prevailing at the moment of shear failure. Thus, for a given normal stress σ, equation (6) correlates the shearing stress τ with the soil properties c and ϕ, and slip j. This equation is empirical and can be determined for each type of soil by finding the appropriate K_1, K_2, ϕ, and c values for any given value of σ.

As an example, consider a firm brittle soil, which may be represented by a hard, well-settled, compact and undisturbed silt. The following K values are likely to obtain: $K_1=1$, and $K_2=1{\cdot}1$. Then, for these values,

$$y_{max}=\left[e^{(-K_2+\sqrt{K_2^2-1})K_1 j}-e^{(-K_2-\sqrt{K_2^2-1})K_1 j}\right]_{max}=0{\cdot}31$$

If the shear test has shown, for instance, that $c=6{\cdot}5$ lb/in² and $\phi=10$ deg, then for $\sigma=3$ lb/in², according to equation (6),

$$\tau_{silt}=\frac{6{\cdot}5+3 \tan 10 \text{ deg}}{0{\cdot}31}\left[e^{-0{\cdot}65j}-e^{-1{\cdot}55j}\right] \quad \ldots\ldots\ldots(7)$$

In the case of a loose sandy loam with a small moisture content, the following K values may represent the stress-strain curve of a shear test: $K_1=0{\cdot}3$, $K_2=2$. If, for instance, the shear test discloses that $c=0{\cdot}2$ and $\phi=33$ deg, then for $\sigma=3$ lb/in², equation (6) may be written analogically in the following form:

$$\tau_{loam}=\frac{0{\cdot}2+3 \tan 33 \text{ deg}}{0{\cdot}76}\left[e^{-0{\cdot}6j}-e^{-1{\cdot}11j}\right] \quad \ldots\ldots\ldots(8)$$

Both curves, τ_{silt} and τ_{loam}, are plotted in Fig. 3. They appear to present a stress-strain relationship that is likely to be encountered in average conditions. Although equation (6) does not include all the shear-strain characteristics of various soils, it may serve as a useful tool in further investigation of the slip-tractive effort relationship of a track moving in a specific soil.

The equation, if written in dimensionless form, also may be helpful in attempts to classify soil types in accordance with the form of their τ (j) curve:

$$\frac{\tau}{c+\sigma \tan \phi}=\frac{e^{(-K_2+\sqrt{K_2^2-1})K_1 j}-e^{(-K_2-\sqrt{K_2^2-1})K_1 j}}{y_{max}} \quad \ldots\ldots(9)$$

The amount of soil distortion, j, which is zero at the beginning and reaches its maximum value, j_m, at the end of the ground contact area, Fig. 2b, can be represented by a triangle, Fig. 4. On the other hand, the slip i_o of a vehicle is expressed by the conventional formula:

$$i_o=1-\frac{v_a}{v_t}$$

where v_a and v_t are the actual and theoretical speeds of locomotion, respectively. The slip and the amount of maximum distortion of the soil j_m can be related as follows. The distance of shear j_m is equal to the speed of slip v_j times the time t in which it occurs:

$$j_m=v_j t$$

Since, however, $v_j=v_t-v_a$,

$$j_m=(v_t-v_a)t$$

On the other hand, $t=s/v_t$, where s is the distance along which j_m has occurred. Hence:

$$j_m=s\left(1-\frac{v_a}{v_t}\right)=i_o s \quad \ldots\ldots\ldots(10)$$

The amount of soil distortion that takes place at any point at a distance x from the front of the ground contact area, Fig. 4, is:

$$j=j_m\frac{x}{s} \quad \ldots\ldots\ldots(11)$$

From equations (10) and (11), it follows that

$$j=i_o x \quad \ldots\ldots\ldots(12)$$

Since the local tangential stress, or the local tractive effort per unit of track area, is determined by equations (6) and (12), the total tractive effort developed by one track is:

$$\frac{1}{2}H=2l\int_0^s \tau\,dx \quad \ldots\ldots\ldots(13)$$

or,

$$H=4l\int_0^s \frac{c+\sigma \tan \phi}{y_{max}}\left[e^{(-K_2+\sqrt{K_2^2-1})K_1 i_o x}-e^{(-K_2-\sqrt{K_2^2-1})K_1 i_o x}\right]dx$$

and

$$H=\frac{4sl\,(c+\sigma \tan \phi)}{K_1 i_o s\, y_{max}}\left[-\frac{1}{-K_2+\sqrt{K_2^2-1}}+\frac{e^{(-K_2+\sqrt{K_2^2-1})K_1 i_o s}}{-K_2+\sqrt{K_2^2-1}}+\frac{1}{-K_2-\sqrt{K_2^2-1}}-\frac{e^{(-K_2-\sqrt{K_2^2-1})K_1 i_o s}}{-K_2-\sqrt{K_2^2-1}}\right] \quad \ldots\ldots\ldots(14)$$

Equation (14) expresses the tractive effort in terms of soil properties c, ϕ, track dimensions $2l$ and s, unit track load $\sigma=W/2(2ls)$, and slip i_o for a given type of soil defined by the constants K_1 and K_2.

The tractive effort per unit track area $4l/s$ was computed in accordance with equation (14) for the previously specified load and silt, and is plotted in Fig. 5. This figure shows the changes of $H/4ls$ with the distortion of soil $j_m=i_o s$, that is, with vehicle slip i_o.

Since $j=i_o x$, it follows from equation (6) that the tractive force τ_x, per unit of the track area, at a distance x from the front of the ground contact surface, Fig. 4, is:

$$\tau_x=\frac{c+\sigma \tan \phi}{y_{max}}\left[e^{(-K_2+\sqrt{K_2^2-1})K_1 i_o x}-e^{(-K_2-\sqrt{K_2^2-1})K_1 i_o x}\right] \quad \ldots\ldots\ldots(15)$$

Thus, it will be seen that the distribution of the elements of the tractive effort along the ground contact area is not uniform and depends on the amount of slip as well as on the properties of the soil.

Curves showing the distribution of τ_x for the previously specified undisturbed soils, which may belong to the categories of firm silts and settled sandy soils, are plotted in Fig. 4, in accordance with equation (15). In firm soils that do not display much strength after rupture, only the frontal portion of the track produces tractive effort, while the rear contributes very little at small rates of slip, $i_o=10$ per cent, and practically nothing at higher slip. The situation is different in soils that do not lose so much of their original strength after distortion beyond the $j_o p_t$ value, Fig. 5. With undisturbed, settled sandy loam, the second half of the track develops a uniform pull for 10 per cent slip. However, even in this case, at 100 per cent slip, almost all the tractive effort is concentrated in the front of the track, with the rear portion practically idling.

The most uniform distribution of the tractive effort, with loose plastic or frictional soil masses, is obtained when their strength is not reduced after the distortion, determined by $j_o p_t$, has taken place. For an average track length s, such soils (dry sand, and supersaturated clay) produce the maximum pull H at 20-40 per cent slip. An unscaled curve of this type is shown by the dotted line in Fig. 5, which illustrates the general shape of a function that is representative of this type of soil. Its shape presumably is determined by $K_1 \approx 0{\cdot}2$ and a value for K_2 of between 3 and 4.

From Fig. 5, an important conclusion can be drawn regarding the length of s of the ground contact area. In this graph, the value $H/4ls$ is plotted as a function of $j_m = i_o s$. If, for instance, a constant s equal to 100 in is assumed, the j_m scale can be written in terms of slip: $i_o = j_m/100$. Such a scale is traced below the j_m abscissa and indicates that the maximum traction is developed in these soils at the following approximate slips:

$s = 100$ in { silt: $i_o s = 2$ or $i_o = 2$ per cent; loam: $i_o s = 5$ or $i_o = 5$ per cent

If, however, the track length s is only 50 in, the maximum traction is obtained at different slip values:

$s = 50$ in { silt: $i_o = (2/50)100 = 4$ per cent; loam: $i_o = (5/50)100 = 10$ per cent

Thus, the amount of slippage depends on the absolute value of the track length, all other variables being equal. In general, then, it may be concluded that a short crawler will slip more than a long one. Since slip is a source of power loss, a long narrow track should be more economical than a short wide one.

The tyre

Consider a very wide, small diameter tyre that would produce the same ground pressure as that of the track. If such a tyre, or a rubber cylinder, is 30 in diameter, the theoretical length s of its ground contact area would be of the order of 5 in. Accordingly, its slip would amount to:

$s = 5$ in { silt: $i_o = (2/5)100 = 40$ per cent; loam: $i_o = (5/5)100 = 100$ per cent

From this example, it can be seen that the slip losses of a vehicle equipped with such a high floatation tyre would be very large in this type of silt soil, and entirely prohibitive in loam. In general, it may be concluded that although it is possible to provide wheels with the same ground contact area as that which can be provided by tracks, that is, with the same floatation, there is inevitably a considerable slippage. This fundamental disadvantage of the wheel, or tyre, as compared with a track may be overcome if the diameter is increased to give a large length s, or by reducing the loads. The analytical method discussed in this paper, and confirmed in its principle by experiment, provides a tool that may be helpful in solution of the track *versus* wheel problem.

ALUMINIUM IN ROAD TRANSPORT*

Present-day Usage and Future Prospects

AT the beginning of the century aluminium, because of its light weight and excellent "workability," was becoming widely used by motor manufacturers. Within twelve years the motor industry became the largest single user of aluminium, and in the 1920's development reached its peak. At that time, the A.C. motor car, typical of its kind, had an aluminium body, and in the engine and transmission many of the heavier castings, notably a one-piece cylinder block and crank case and a combined rear axle and gear box, were in aluminium. Later, with the introduction of the popular car by Ford, effort was concentrated upon the development of a new technique of production rather than upon materials.

The influence of flow-line production methods on design has led to the present relatively light weight, high performance vehicles made largely of ferrous materials. During the last few years, however, the need to increase top gear performance, acceleration, braking and road holding, has created interest in further reduction of weight, and is leading once more to an increased use of aluminium.

Against such a background, briefly sketched, the use of aluminium in some modern vehicles can be set with clearer significance, though, for the most part, with equal brevity. In the Rover 75 Saloon, for example, approximately 96 lb of aluminium are used in the bonnet, doors and boot lid, resulting in a total weight saving of 120 lb. The Austin Princess departs from general practice in its use of pressed aluminium panelling in place of the hand beaten panels usually fitted to high quality cars. Another excellent example is the latest Park Ward body, usually mounted on a Bentley chassis: the use of aluminium here extends to the actual framing members, which are folded up from sheet material and assembled by argon arc welding or riveting. Tests over two years give every reason for confidence in this design. One of the most outstanding lightweight, series produced cars is the Panhard Dyna. Here, by the extensive use of aluminium, in body and engine, a kerb weight of 13¾ cwt has been achieved for an "occasional six-seater" car having a very high performance; 80 m.p.h. with 41·2 brake horse power, and a claimed fuel consumption of 40 m.p.g.

There is no doubt that the extensive use of aluminium to replace steel and cast iron for both the body and the engine of the private car could reduce weight by 20 per cent. A car with a kerb weight of one ton should, in aluminium, weigh only 16 cwt. for an increased material cost not exceeding £30. It has been estimated that the percentage decrease in weight roughly equals the percentage increase in miles per gallon, provided engine power is adjusted to give equivalent performance. In cars of traditional form it is unlikely that gross weight saving, and hence the mileage per gallon, will exceed 16 per cent. Nevertheless, for an assumed original 30 m.p.g., 5 gallons will be saved per 1,000 miles, which means, at to-day's prices, that the additional material costs would be written off by fuel savings alone in 26,000 miles.

Experience with aluminium wheels on competition cars indicates that their use is likely to be extended and will, of course, lead to a reduction of unsprung weight. The Cadillac Company are introducing wire wheels with forged aluminium rims for their 1955 models. These are produced to very close tolerances, and the true-running characteristics improve tyre life. The speed and cornering power of present-day cars generate much heavier loads, particularly on the wheel disc, but, nevertheless, tests have been carried out on vehicles with aluminium centres using an aluminium-5 per cent magnesium alloy in direct replacement of steel, which shows a fatigue life sufficiently high to suggest that the aluminium wheel is a distinct possibility. If further reductions in unsprung weight are to be made, aluminium will have to be considered for disc wheels in order to maintain sufficient lateral stiffness.

Aluminium trim is expected to become increasingly popular on cars. This material consists of a very high purity aluminium alloy, electrolytically or chemically brightened and afterwards anodized. Coaches with aluminium trim have now been running for six months and have demonstrated its remarkably good appearance in service. Aluminium trim costs about the same as steel, chromium plated, and much less than chromium plated brass.

Commercial transport operators have very definite views on the cost of saving weight. In passenger service vehicles, expenditure on fuel, tyres and repairs is closely related to unladen weight and largely governs the cost per mile. It has been estimated that an expenditure of 3s. 6d. can be incurred for a weight saving of 1 lb; when aluminium is used, it is unlikely that the cost of weight saving will exceed 2s. 3d. per lb. In all forms of transport, improved methods of design result in reductions in material thickness, but with aluminium the limit is set by considerations of appearance rather than corrosion. On many designs, outside truss panels could be reduced in thickness if stiffening members were added between the pillars. Resin bonding provides an ideal method of attaching interior stiffeners without affecting outside finish in any way.

*From a paper presented by T. G. Griffiths, B.Sc. (Eng.), A.F.R.Ae.S., *and* D. R. Thomas, A.M.I. Mech.E., *at the Aluminium Centenary Exhibition.*

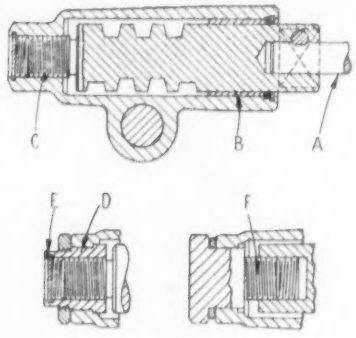
CURRENT PATENTS

A Review of Recent Automobile Specifications

Mounting of steering gear cam

IT is proposed to mount the cam of a steering gear in a plain bearing and a screw-thread type bearing instead of the more usual ball bearings. Construction is thereby simplified and cheapened and an inherent means of adjusting the axial position of the cam is provided.

The end of the cam adapted for connection to the steering shaft A is supported in the gear casing by a plain bush B furnished with suitable sealing and retaining rings. At its opposite end the cam is formed with an axial extension C of reduced diameter which is externally screw threaded to take a bearing in a screw-threaded bore in the end of the casing. The threads are of V-type and of fine pitch. Rotation of the cam during operation will cause a slight axial displacement in one or the other direction according to the hand



No. 717523

of the screw thread, but this has no detrimental effect and can, if desired, be compensated for in the design of the cam. It is preferable for the screw thread and the cam groove to be of opposite hand. The axial position of the cam relative to other elements of the gear can be adjusted by varying the extent to which the extension C is engaged in the screwed bore.

In a modified arrangement the screw-thread bearing is constituted by a bush D rotatably mounted in the casing and secured by a lock nut. Rotation of this bush by means of spanner flats at E, provides a convenient method of adjusting the axial position of the cam. Alternatively, the cam may have a screwed bore engaged on a screwed plug F which is "suitably anchored" in the casing. *Patent No.* 717523. *R. Bishop, R. H. Johnston* and *G. B. R. Whitlock.*

Automatically adjustable valve timing

IN general practice a fixed valve timing is determined as a compromise to meet the requirements of varying engine speeds and loadings. It is possible to design an engine to give high power at maximum speed but, in that case, torque and specific fuel consumption will be less than the optimum in the medium speed range. Conversely, favourable torque and consumption values can be obtained at moderate speeds, but with a relatively lower output at maximum speed. Similarly, though to a lesser degree, optimum valve timings can be allocated to different engine loadings.

The invention is particularly concerned with inlet valve control, with the object of obtaining as full a charge as possible. At low engine speeds admission should be terminated relatively early to obviate the danger of a return flow to the manifold. When engine speed is increased, the cut-off should be later in order to take advantage of the inertia effect of the entering column of air or mixture. In a diesel engine, for example, the most favourable cut-off points were 30 deg past B.D.C. at 1,000 r.p.m., and 55 deg past B.D.C. at 3,000 r.p.m.

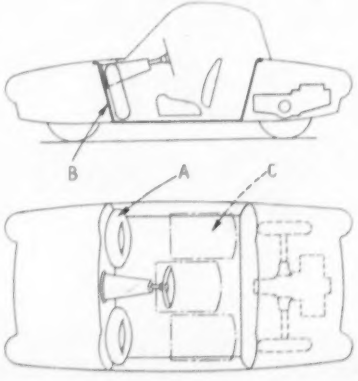
Accordingly, it is proposed to vary valve timing or period of valve opening during operation by angular or axial displacement of a suitable camshaft. Control is to be effected, directly or indirectly, by a speed-responsive governor as the main factor and a capsule influenced by pressure in the intake manifold as a correctional factor. Such a dual control may be arranged by means of a two-armed lever working on an adjustable pivot.

The specification is framed in general terms and no drawings are included. *Patent No.* 716997. *Daimler-Benz A.G. (Germany).*

Vehicle construction

DISPOSITION of a pair of spare wheels in the main body space is the feature of this invention. In either a front-engined or a rear-engined vehicle they are arranged substantially vertically on the bulkhead remote from the engine in order to secure a distribution of weight. To prevent the odour of rubber from pervading the passenger space the wheels may be housed in a sealed compartment, access to which is by apertures in the side of the body closed either by the vehicle doors or by separate flaps.

The illustration shows a three-seater, rear-engined vehicle with a central driving position. Spare wheels A are mounted on



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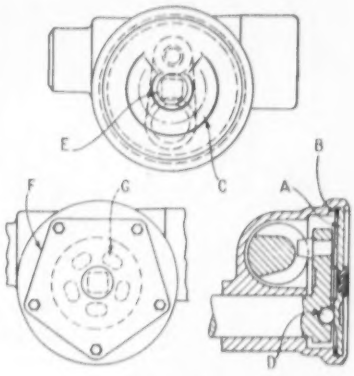
the front bulkhead B, one on each side of the steering column. Preferably the two side seats C are set back in relation to the driving seat.

Alternative arrangements for front-engined, front-wheel driven vehicles are described in the specification. One has the two spare wheels carried coaxially on the rear bulkhead in a central position to the rear of the driving seat. They are flanked on each side by luggage compartments, and on the outer side of the bulkhead is mounted the centrally disposed fuel tank.

In the second layout the rear bulkhead is recessed inwardly to receive the externally mounted central fuel tank and the spare wheels are accommodated in side compartments. *Patent No.* 717069. *Daimler-Benz A.G. (Germany).*



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Steering gear thrust adjustment

IN this cam-type steering gear the thrust of the rocker shaft is taken directly by the cover plate which is rotatably adjustable to position the rocker shaft axially. The sheet metal cover plate A is seated on a rubber ring B in a recess in the gear housing and retained by spinning over the edge of the housing. Alternatively, it may be secured by a spring ring or by a series of bridge-type clamps and set screws.

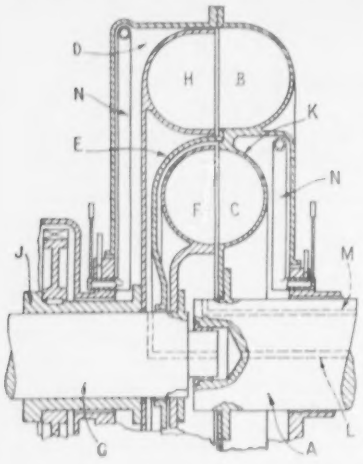
On its inner surface, plate A is pressed out with a part-circular cam formation C rising progressively from one end to the other. Its medial line is an arc concentric with the plate and of a radius equal to the distance from the centre of the plate to the axis of the rocker shaft, where it engages the usual hardened stud or ball D. To facilitate angular adjustment the cover plate is punched out centrally with a square aperture which is closed by a rubber plug E.

In a modified construction a cover plate F, of regular polygonal or circular shape, is secured by means of symmetrically disposed set screws. It may have the continuous cam, as in the previous design, or be formed with a series of abutments G of progressively increasing height, to give a stepped adjustment by turning the plate on the studs. *Patent No.* 717524. *R. Bishop, R. H. Johnston and G. B. R. Whitlock.*

Reversible hydraulic coupling

IN this transmission mechanism two fluid couplings of the fillable type are employed and reversal of the drive is obtained by filling the one and emptying the other and *vice versa*. The two component units are disposed co-axially in a common plane, with the outer one arranged to provide a drive in the more important, usually forward, direction. Without impairing efficiency the inner diameter of the working chamber of a fluid coupling can be substantially increased if the outer diameter is only slightly increased. Accordingly the ratio of inner to outer diameter of the outer coupling is higher than is usual, being not less than 0·5:1 and preferably 0·7:1.

The driving shaft A is flanged to carry a shell furnished with outer impeller blades B and inner blades C and forming the scoop chamber for the inner unit. To this shell is attached the outer unit scoop chamber D and the casing E enclosing the inner runner F mounted on the driven



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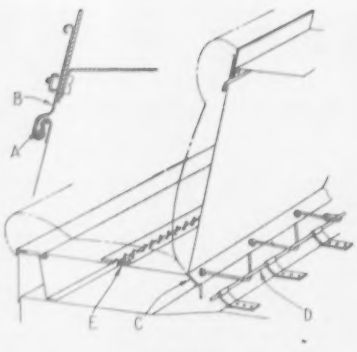
reverse shaft G. Outer runner H is secured to the hollow driven forward shaft J.

Scoop chamber D communicates with its working chamber by way of the clearance around the periphery of runner H, and the other scoop chamber with the working chamber of the inner unit by means of holes K in the impeller. Selective filling of the working chambers is effected by a suitable pump by way of drive shaft drillings L for the inner unit and M for the outer unit, and scoop tubes N are provided for emptying. *Patent No.* 714011. *Hydraulic Coupling Patents Ltd.*

Seat construction

SEAT and back rest portions of this seat are constituted by a single sheet of canvas or similar flexible material, resiliently tensioned from a rearwardly extending fold at the junction of the two portions. The free edges of both seat and back portions are folded and secured to form a transverse loop, in which is inserted a rod A adapted to be fitted into a recess in plate B, which is spot-welded or bolted to the metal framing. A fold C at the junction of the two portions is eyeletted and laced by elastic cord to a transverse rod D bracketed from the framing. The rod is preferably slidable in the brackets to facilitate assembly.



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As alternatives, the elastic cord may be looped over hooks or the cord may be replaced by a series of tension springs, engaged over a rod inserted in the fold C and anchored to frame brackets. To improve the flexibility of either the seat or back portion, it may be box-folded transversely, as at E, eyeletted and laced by elastic cord.

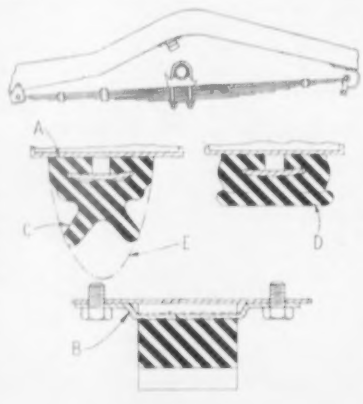
It is claimed that this hammock-like arrangement, being resiliently anchored and tensioned, is comfortably adaptable to occupants of different size and weight. Further, that as the sheet is capable of ready flexing movement, the occupants can obtain maximum comfort and support in various positions. *Patent No.* 717864. *Briggs Motor Bodies Ltd.*

Suspension bumper

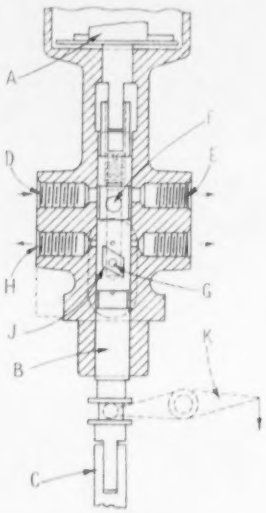
THIS rubber auxiliary bumper is formed with two narrow-section wings projecting from the base. On impact the wings are folded into engagement with the base to form a resilient bumper having a higher rate. In the free state the bumper comprises an elongated block of rubber or other resilient material, having a flat bearing surface A and formed with a T-shaped slot for engagement on the bracket B, by which it is secured to the chassis frame. Both the slot and the bracket are slightly curved transversely to ensure a secure clamping effect and to prevent the spreading of the base under pressure.

The wings C in the example illustrated are arranged at an included angle of 70 deg. As the axle rises on the bounce it first contacts the wings, then spreads them apart and finally folds them down on to the body of the bumper, as shown at D. While initially the rate of the bumper is relatively low, it is greatly accelerated during its second stage. This low initial rate softens the impact and results in quieter operation. Since the low rate results in a greater absorption of energy, the energy returned on the rebound is less than that occurring with a conventional elliptical bumper (indicated at E) and the ride is thereby improved.

Additionally, it is claimed that the wing type bumper is substantially unaffected by temperature changes, whereas the rate of a conventional bumper may increase by from 10 to 30 per cent if subjected to a temperature drop from 70 deg F to 10 deg F. It can be manufactured economically by extrusion and cut to length, instead of by moulding in the usual manner. *Patent No.* 716767. *Ford Motor Co. Ltd.*



No. 716767

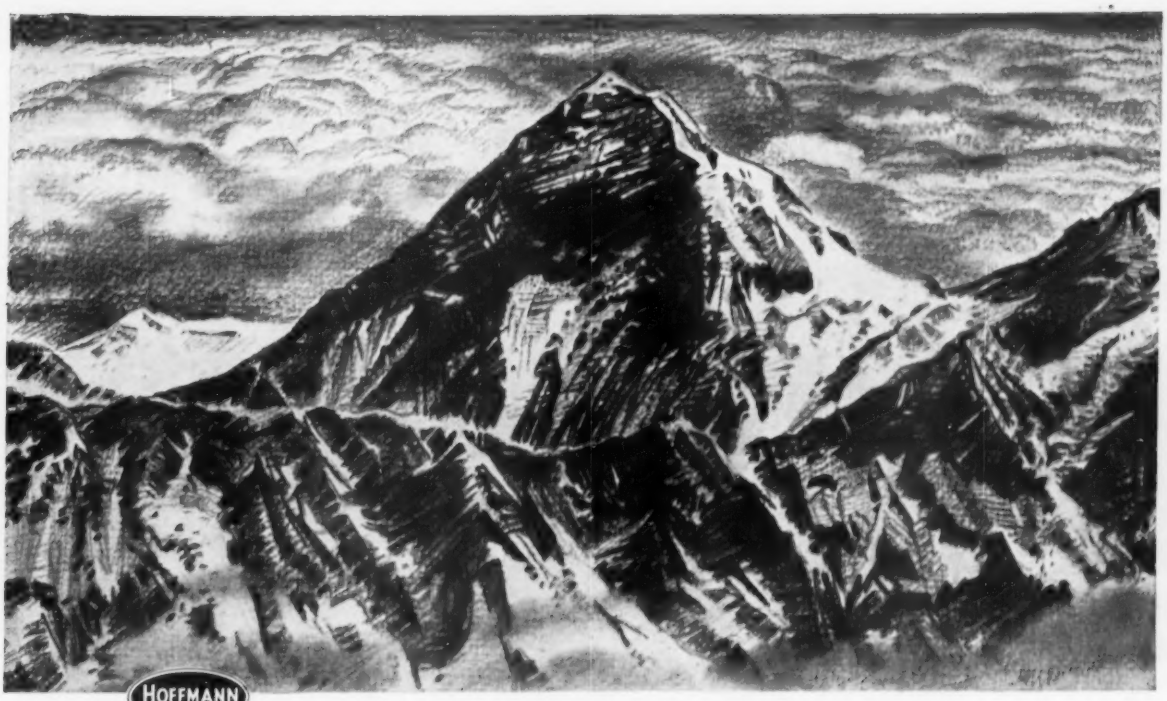


No. 716730

Fuel distributor for petrol-injection engine

IN view of the definite timed relationship between the termination of injection and the point of ignition, it is proposed to combine the fuel and ignition distributors in a single unit with a common drive. This unit comprises an elongated body having an upper chamber in which is accommodated a normal ignition distributor A. The fuel distributor shaft B connects the driving shaft C and the ignition distributor spindle, long tongue-and-groove couplings permitting axial adjustment of position. Fuel under pump pressure is admitted to the body by way of port D and can be returned to the pump through port E under control of a by-pass valve (not shown).

Shaft B has an axial boring, closed by a screwed plug, communicating with an annular recess in the shaft by a radial drilling F. Through a second radial drilling G fuel is passed sequentially to discharge ports H and piped to the nozzles on the engine. Regulation of the quantity of fuel delivered is by means of a control groove J in the form of an oblique parallelogram. As the shaft B is raised by lever mechanism K, injection commences earlier but the cut-off is unaffected. *Patent No.* 716730. *Ustav Motorovych Vozidel, J. Bottger and O. Koplik (Czechoslovakia).*



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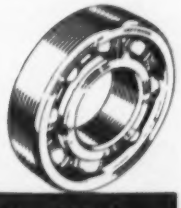

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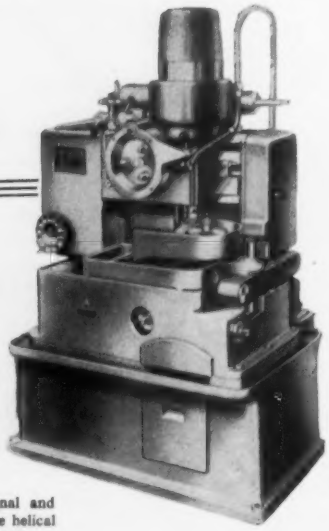

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The photograph shows the actual instrument used by Ford Motor Company Limited at the Motor Industry Research Association's proving ground, Nuneaton. It successfully recorded data while going round the track at 90 m.p.h.

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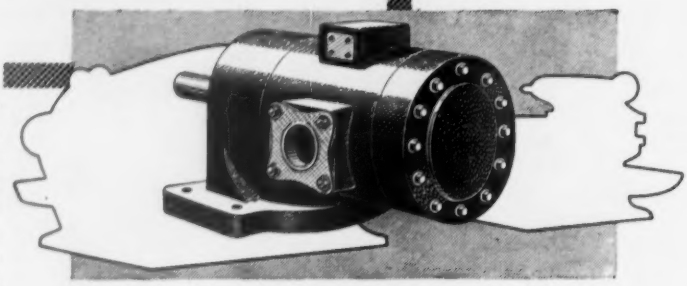

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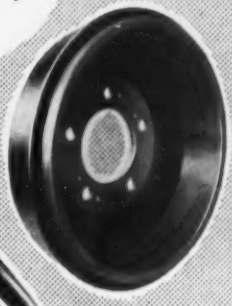

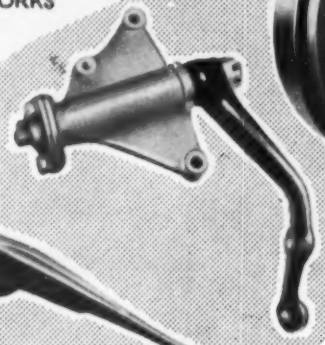

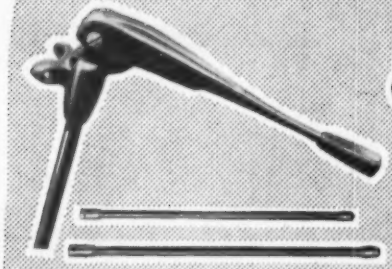

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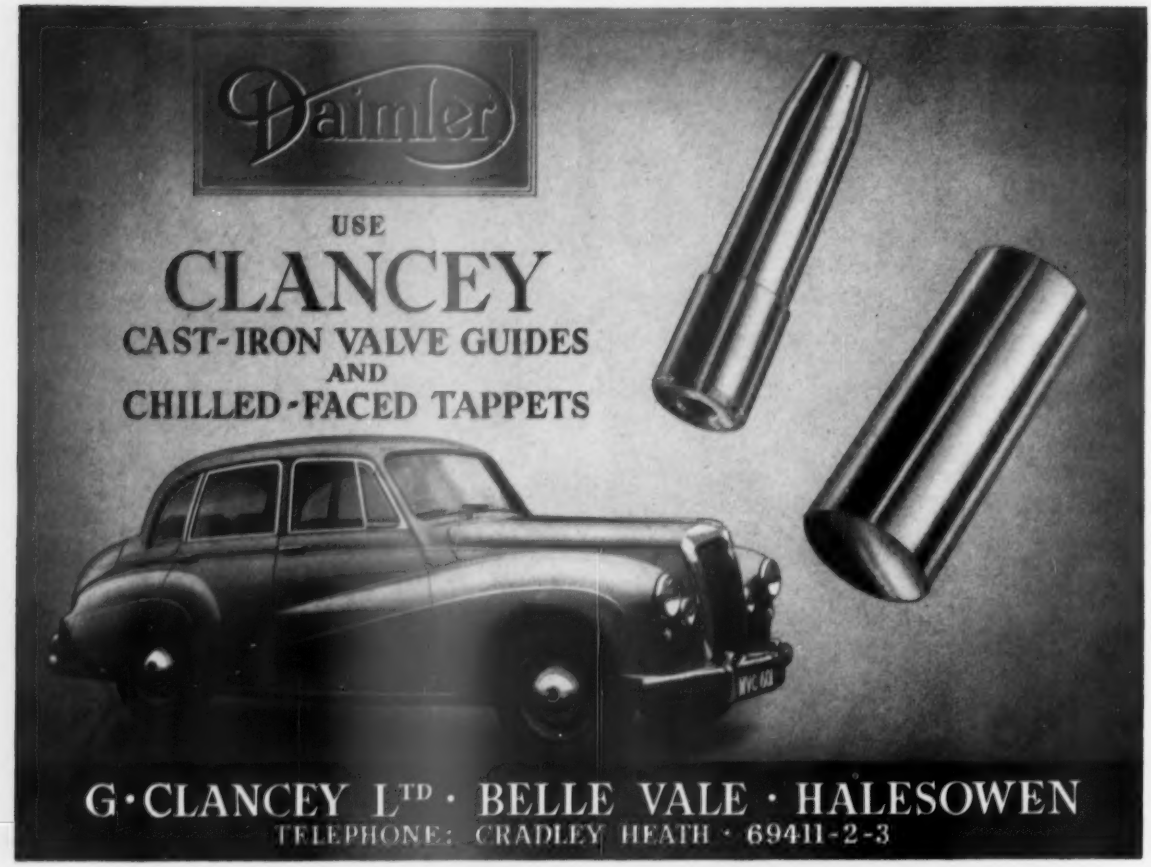

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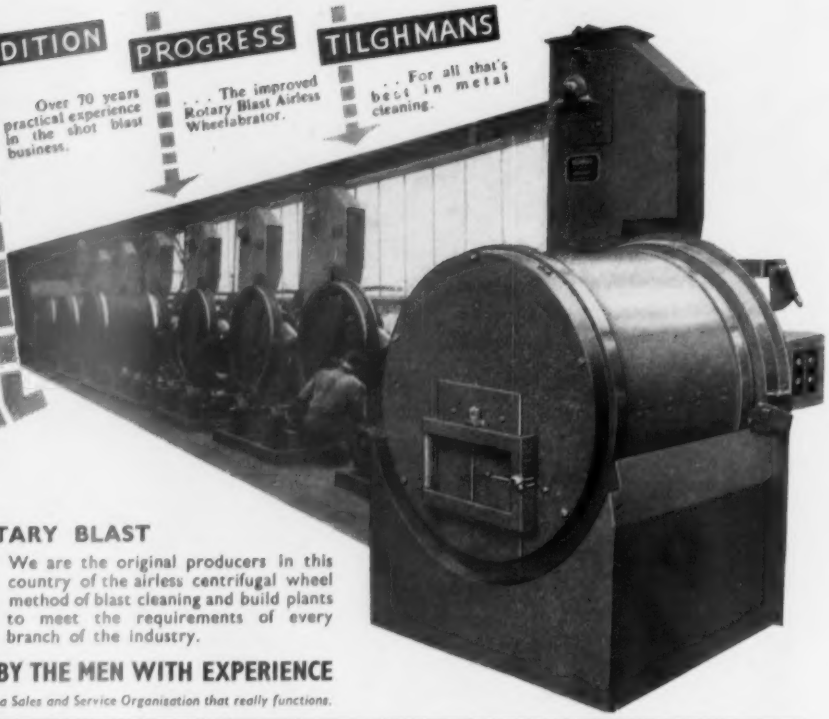

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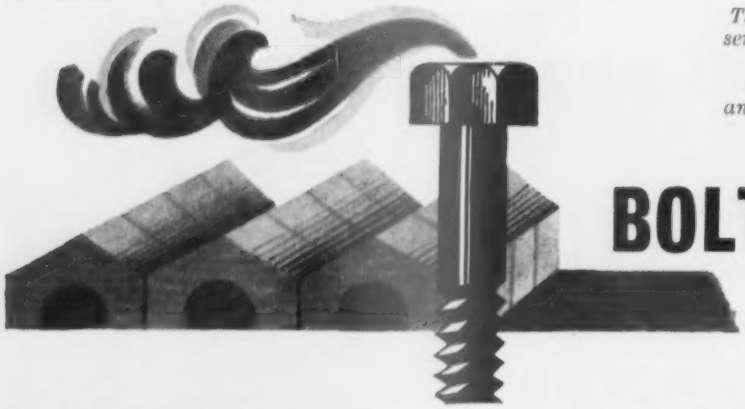

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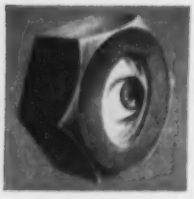

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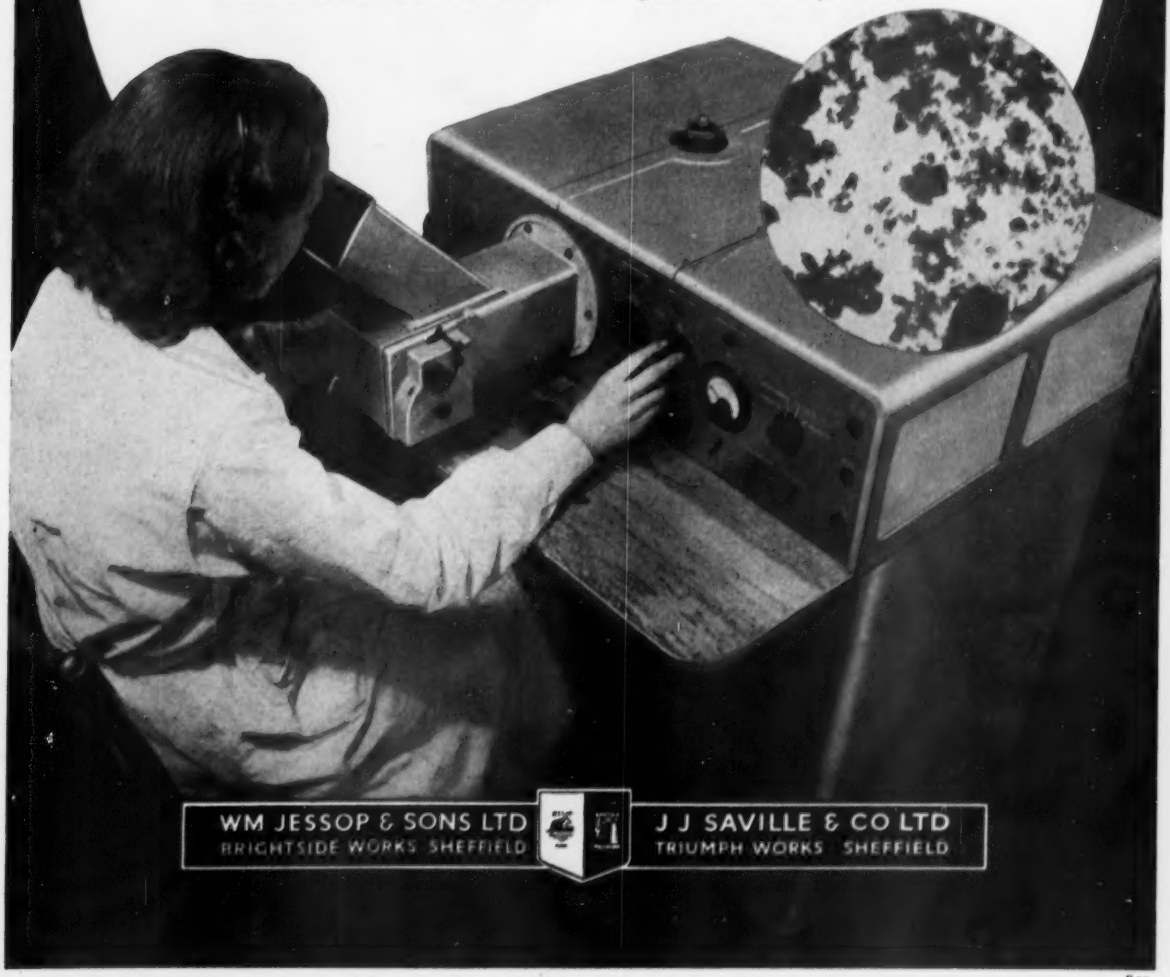

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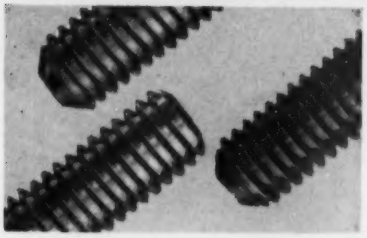

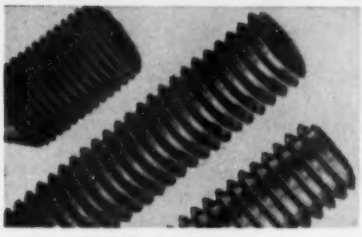

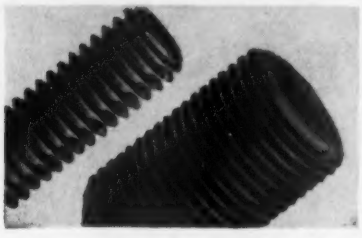

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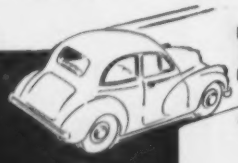

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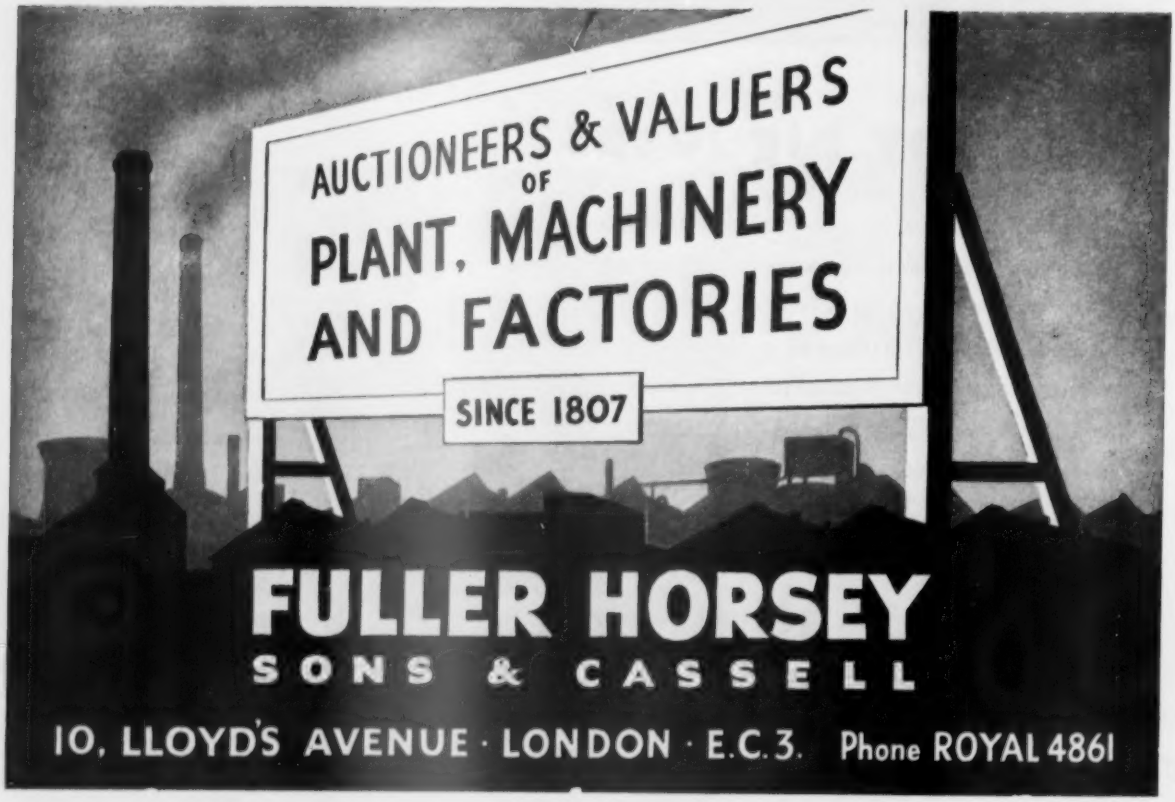

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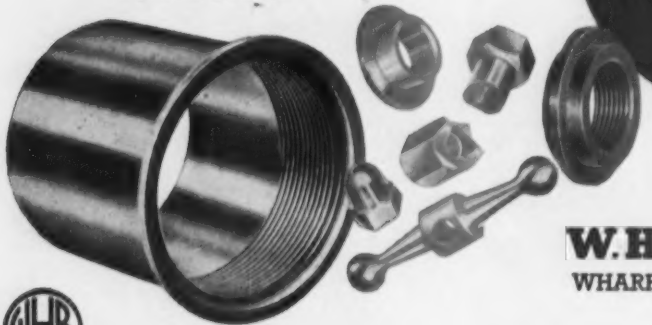

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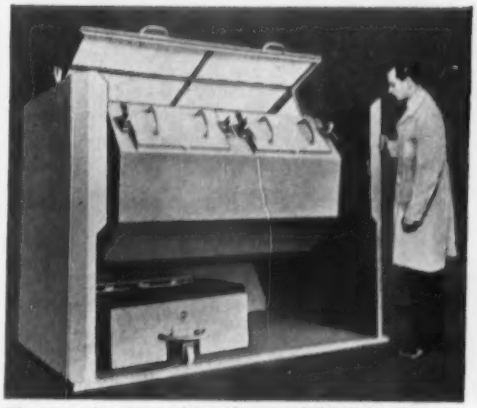

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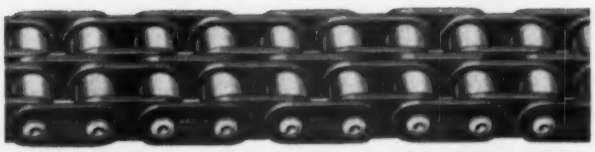

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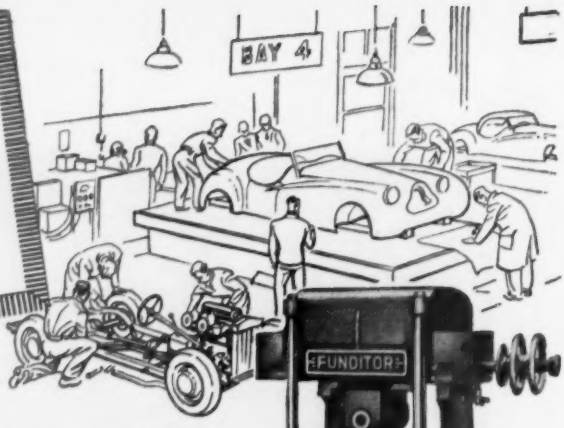

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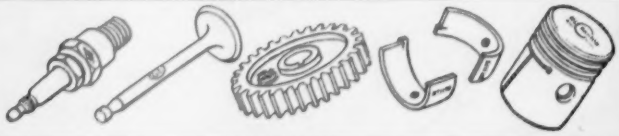

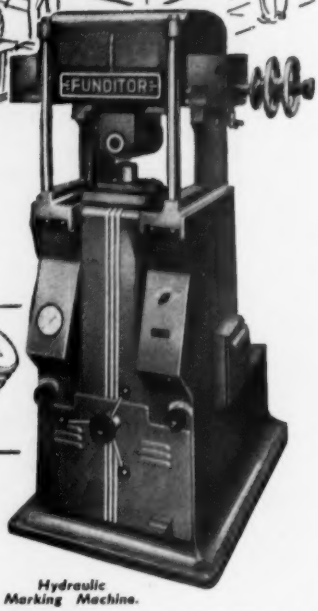

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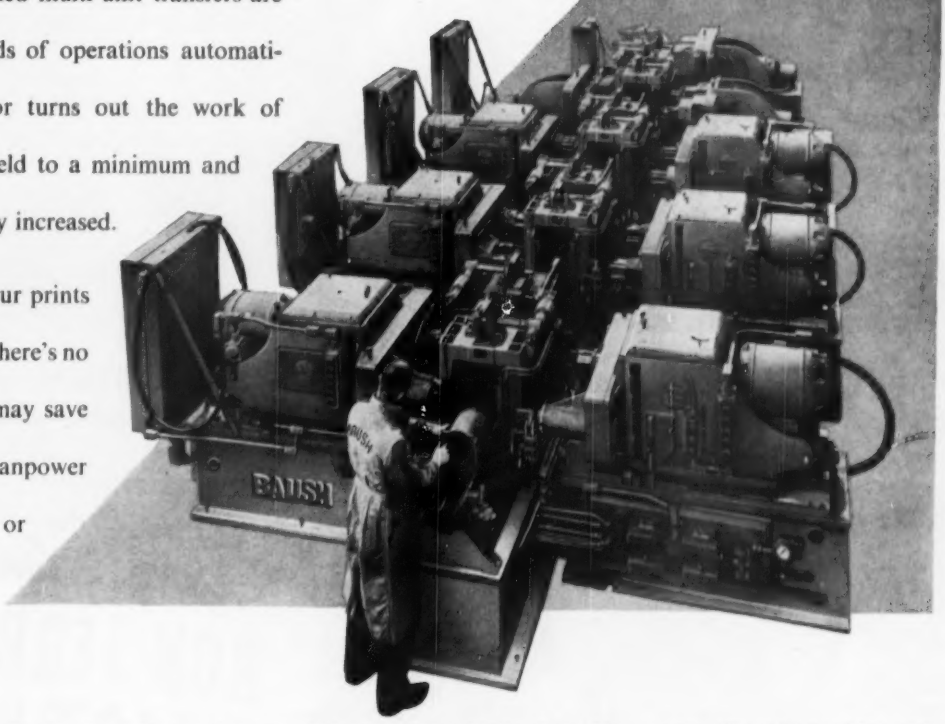

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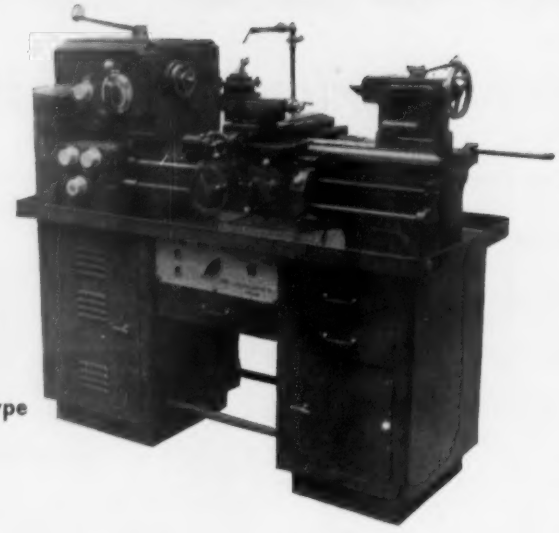

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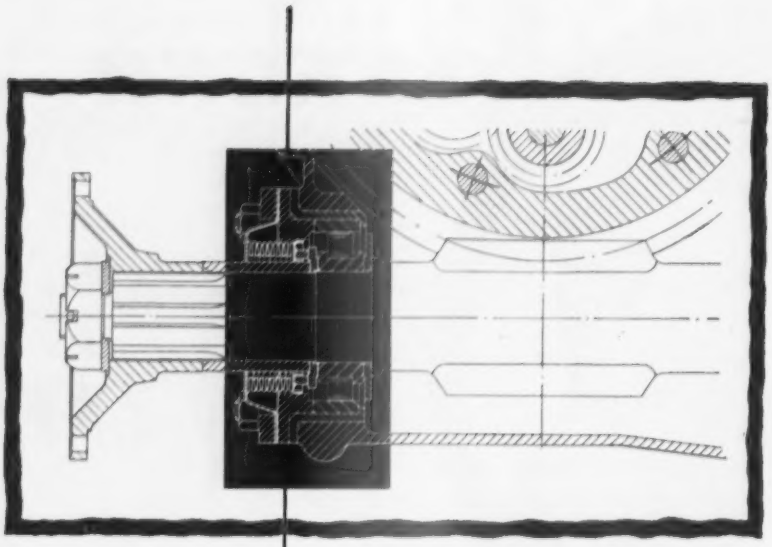

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AFTER several years' intensive investigation, the Rolls-Royce Research Laboratories developed a compound which would effectively seal the crankcase, sump and other engine joints subject to oil and petrol leaks.

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Will not flake, crumble or dissolve. Wellseal will not flake, powder, harden or dissolve in petrol, paraffin, diesel, lubricating oils, water, etc. Wellseal dries to a spongy film, never becomes brittle. It is this plasticity which makes it so suitable for joints liable to fretting, as the resilience of the compound accommodates itself to the slight joint face movement. When necessary joint faces can be parted easily without damage.

Easy to use. Wellseal is easy to use. It can be applied either direct from the tube, or with a brush. It spreads easily, so that every part of the surface is covered. When necessary, it can be quickly removed with a rag dipped in Carbon Tetrachloride or Trichlorethylene.

Wide range of uses. Wellseal is not only the perfect jointing compound for crankcase joints, but is ideal for axle boxes, gear boxes, cover plates, and all oil or water retaining joints.

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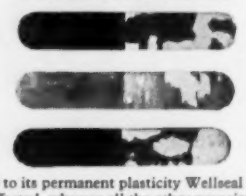
Four strips of tin were painted, one with Wellseal and the others with proprietary jointing compounds. After standing 24 hours the strips were bent double and then straightened.

BEFORE BENDING | AFTER BENDING

WELLSEAL

OTHER COMPOUNDS



Due to its permanent plasticity Wellseal was unaffected, whereas all the other proprietary compounds cracked and flaked.

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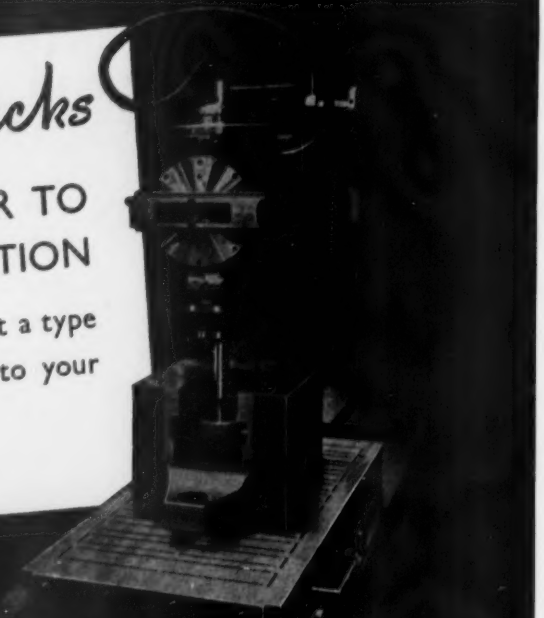

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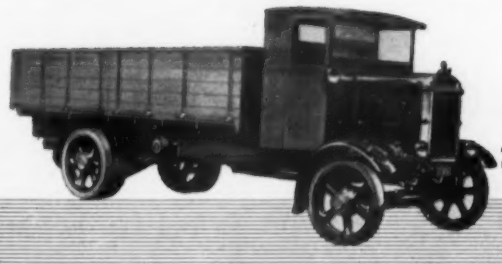

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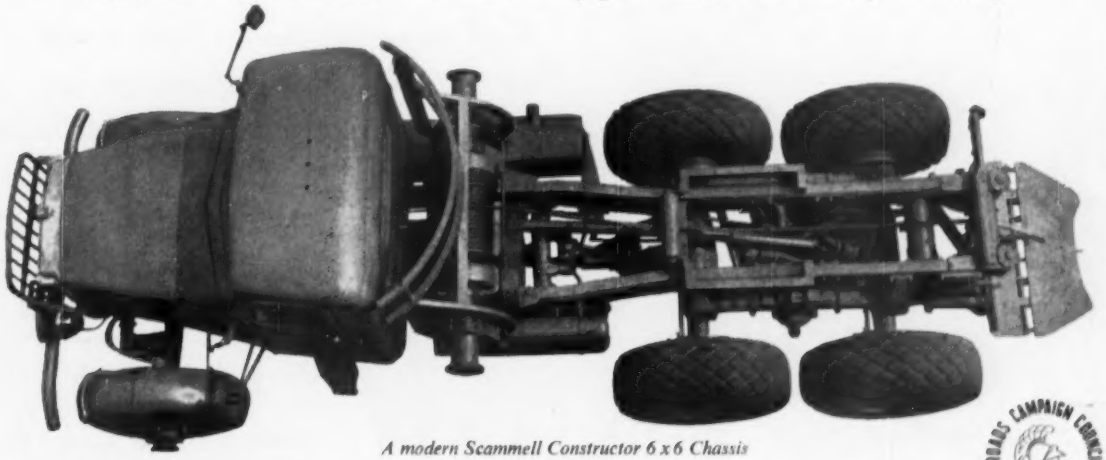

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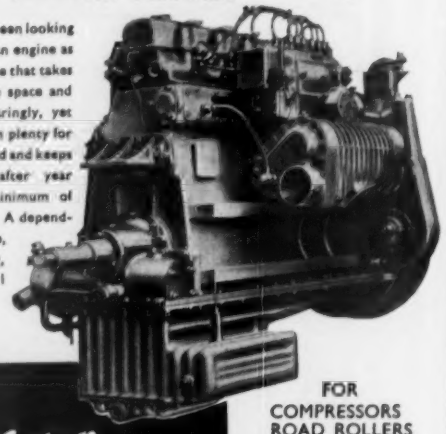

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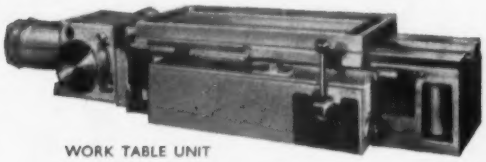

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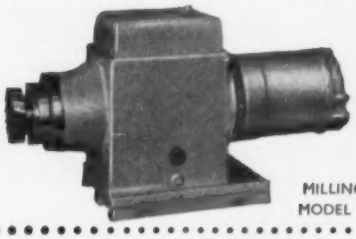

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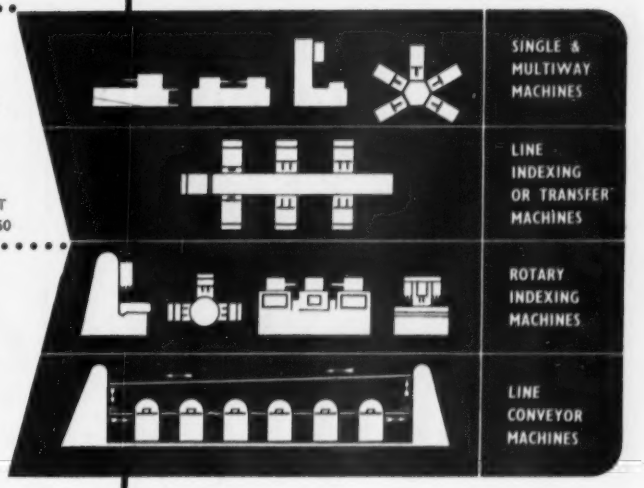
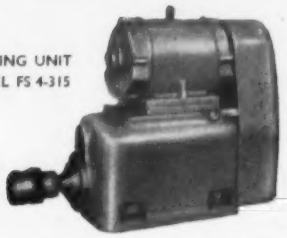
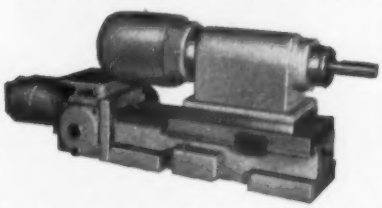




DRILLING UNIT
MODEL FS 4-315



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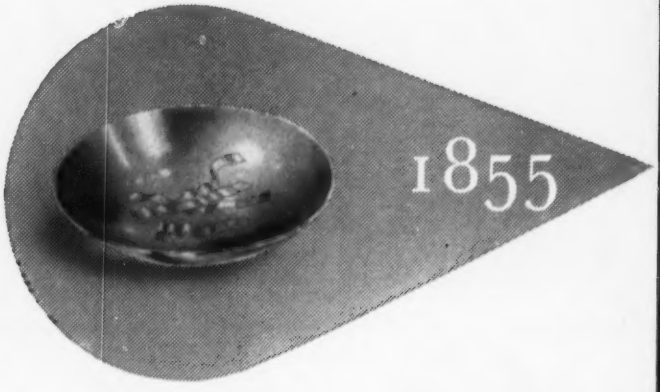

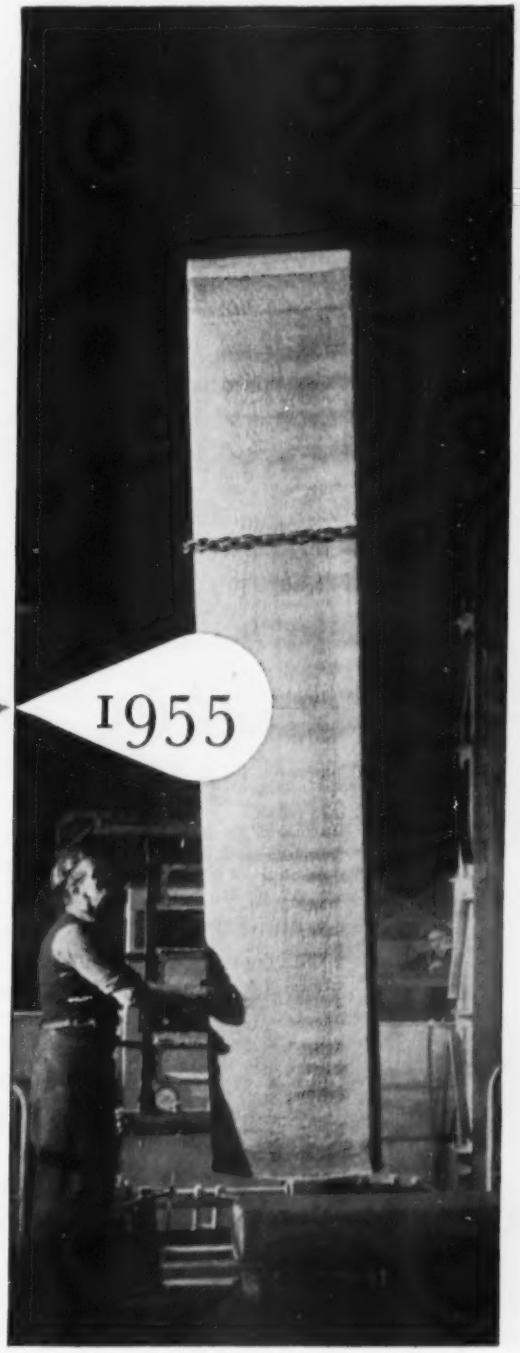

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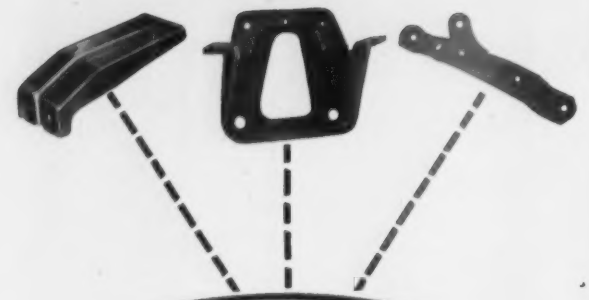

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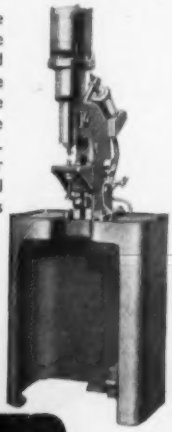

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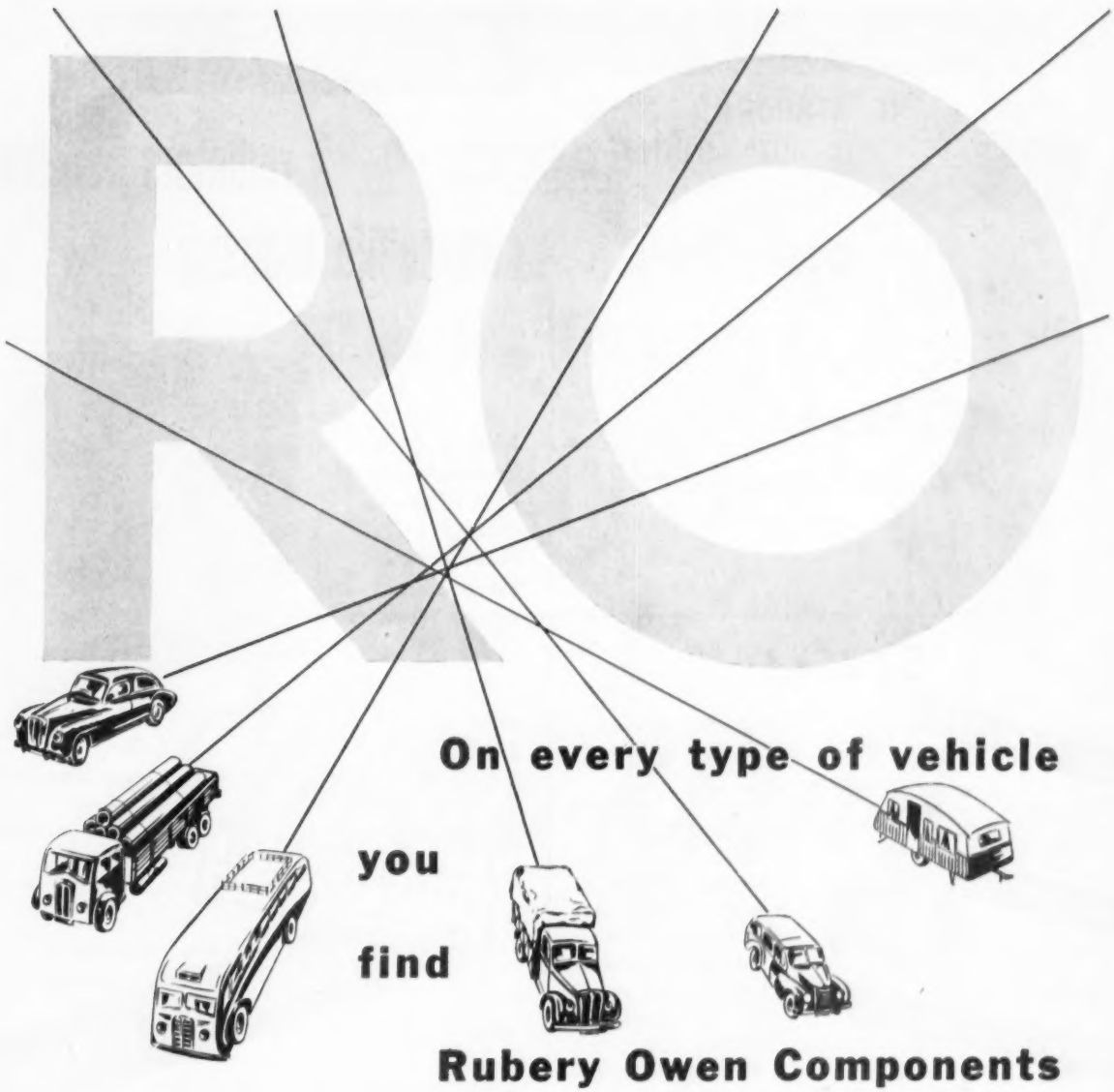

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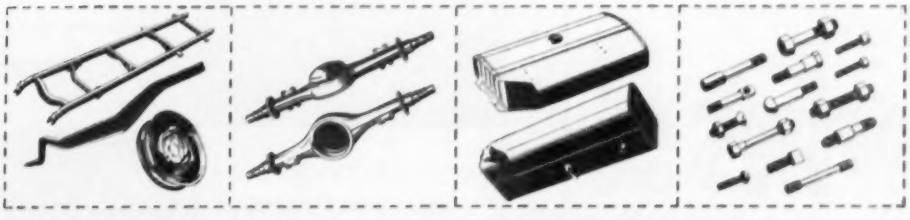

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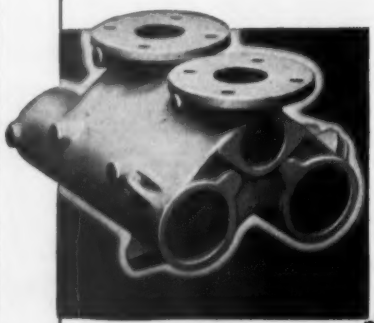

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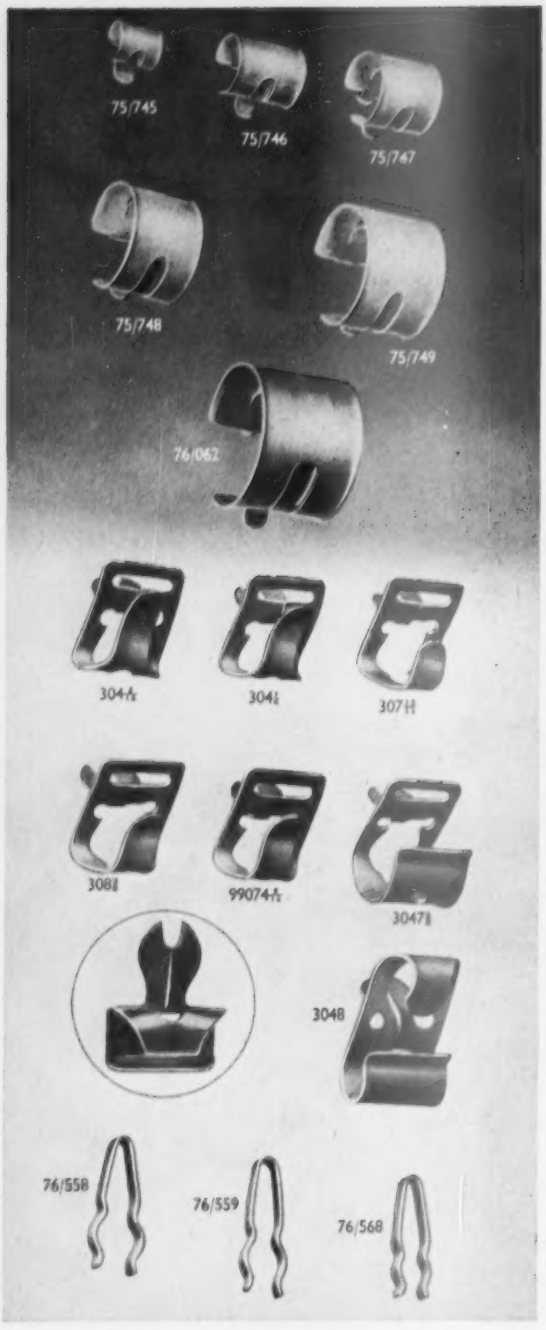

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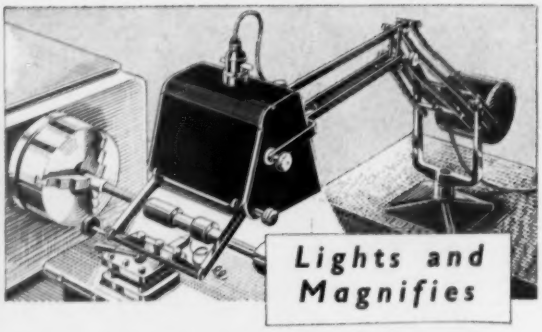

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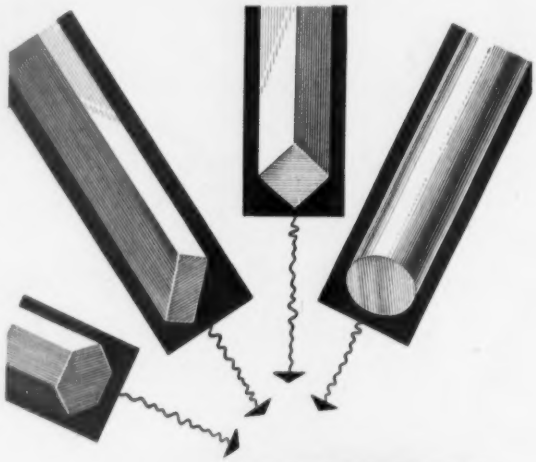

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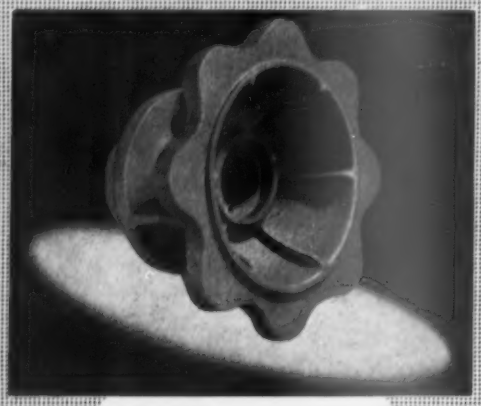

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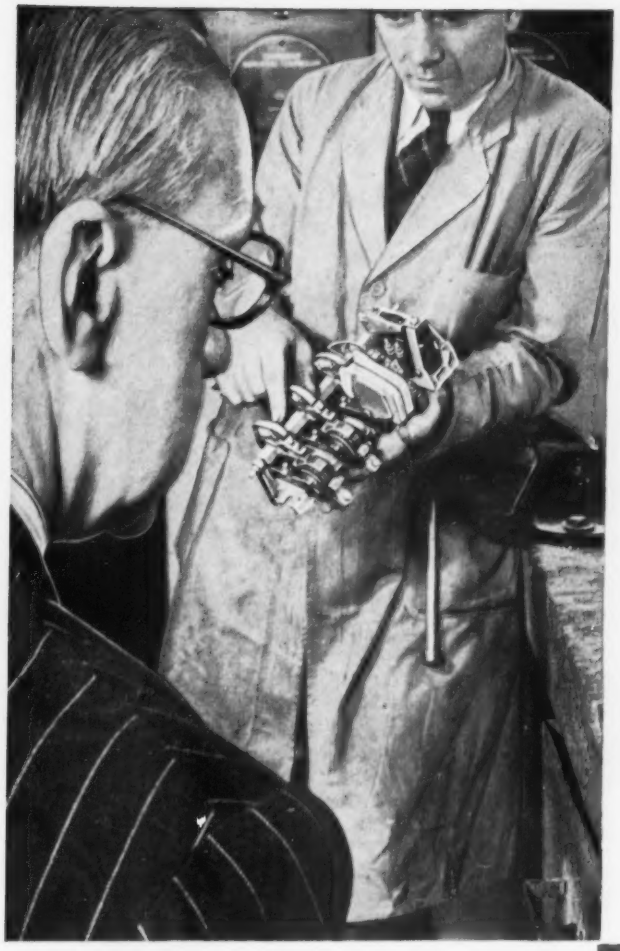

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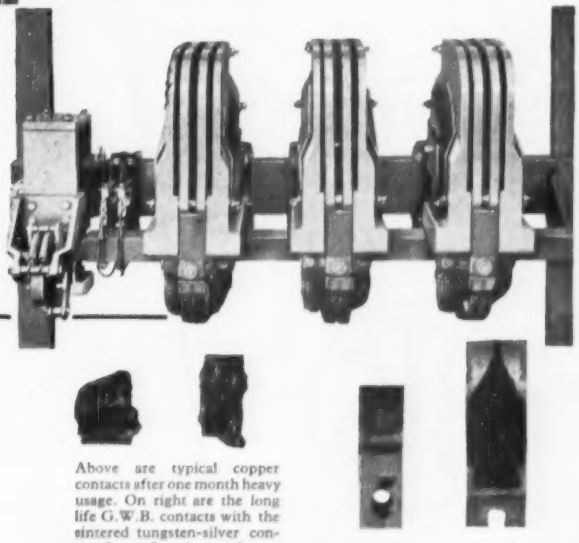

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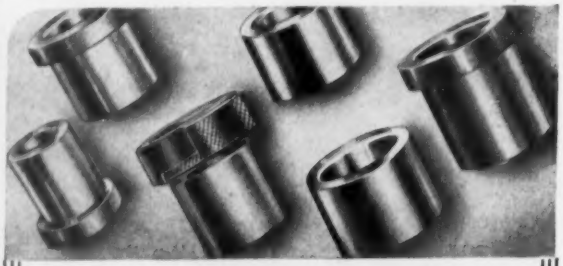

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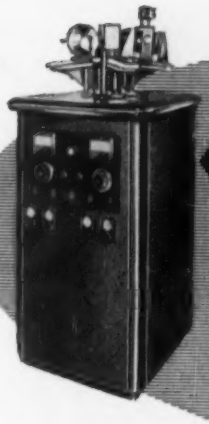

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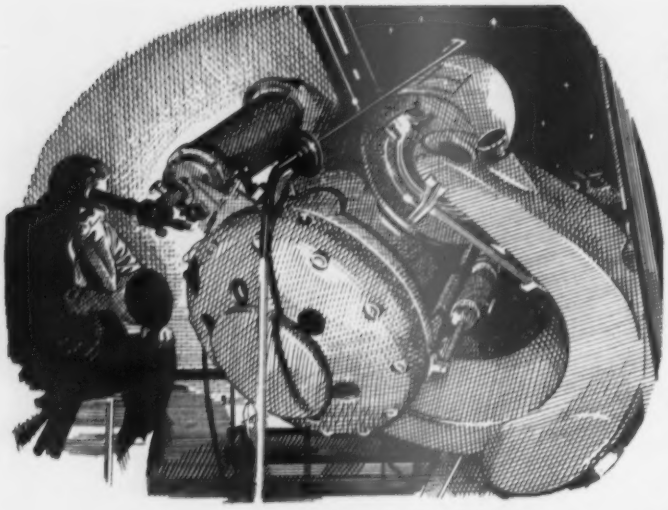

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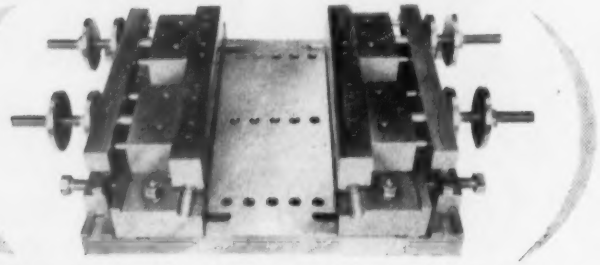

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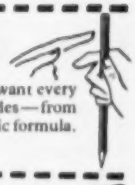

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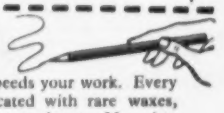

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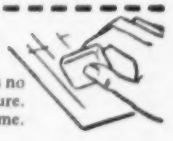

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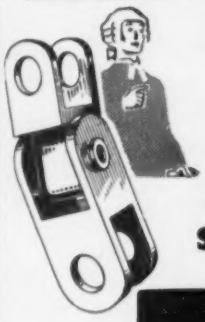

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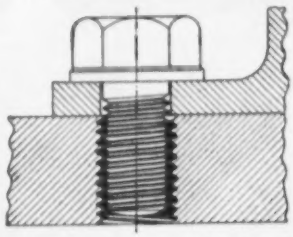

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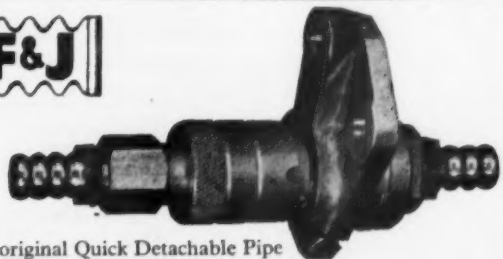

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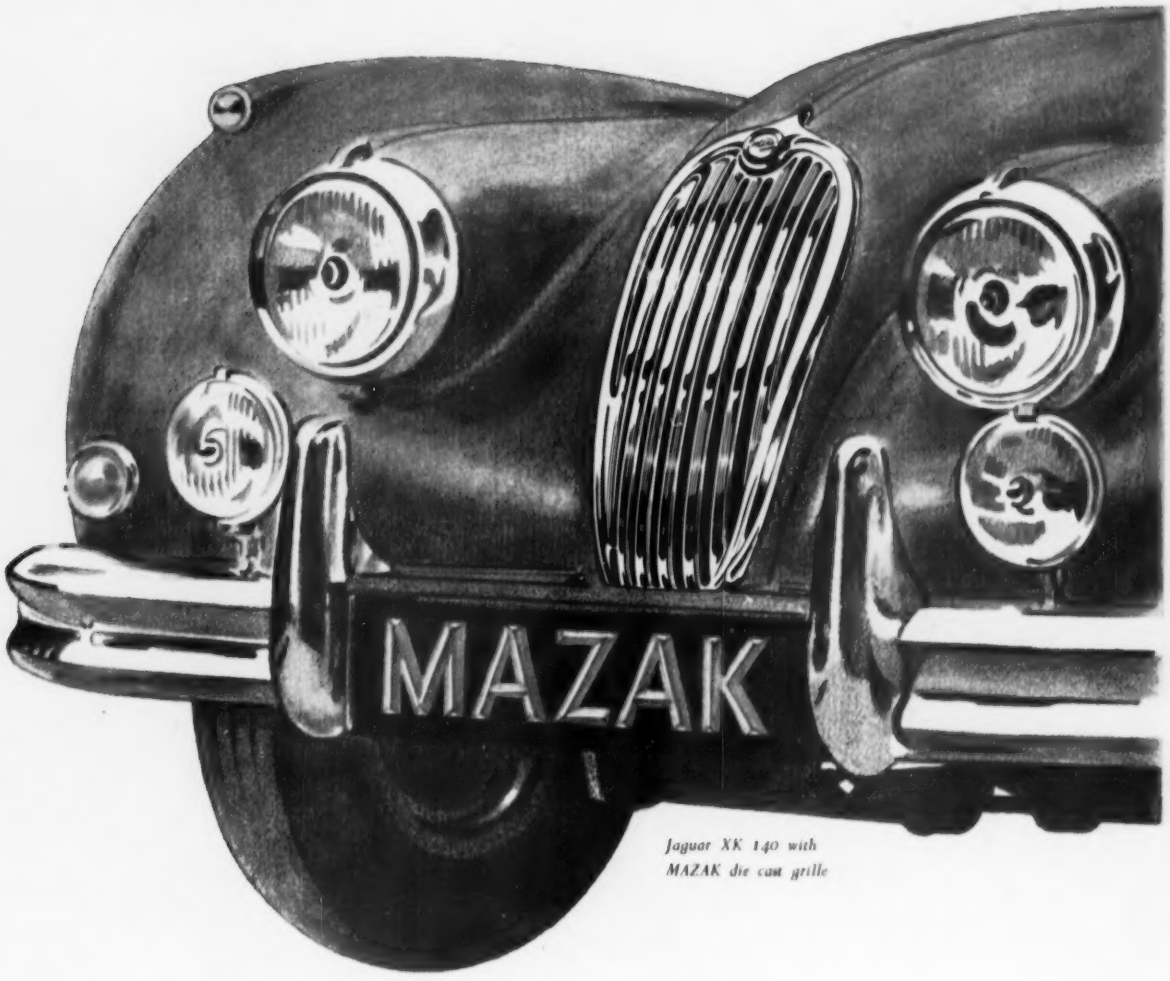

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